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Genter et al.

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(54) **APPARATUS AND METHOD FOR ADJUSTING A GEAR**

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(*) Notice: Subject to any disclaimer, the term of this
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This patent is subject to a terminal dis-
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Primary Examiner—Allan D. Herrmann

(22) Filed: **Jan. 21, 2000**

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(63) Continuation of application No. 09/186,238, filed on Nov. 4,
1998, now Pat. No. 6,109,129, which is a continuation-in-
part of application No. 08/853,341, filed on May 8, 1997,
now Pat. No. 5,870,928, and a continuation-in-part of appli-
cation No. 08/853,013, filed on May 8, 1997, now Pat. No.
5,979,259.

(57) **ABSTRACT**

(51) **Int. Cl.**⁷ **F16H 55/18**

(52) **U.S. Cl.** **74/440; 74/397; 74/409**

(58) **Field of Search** **74/397, 409, 440**

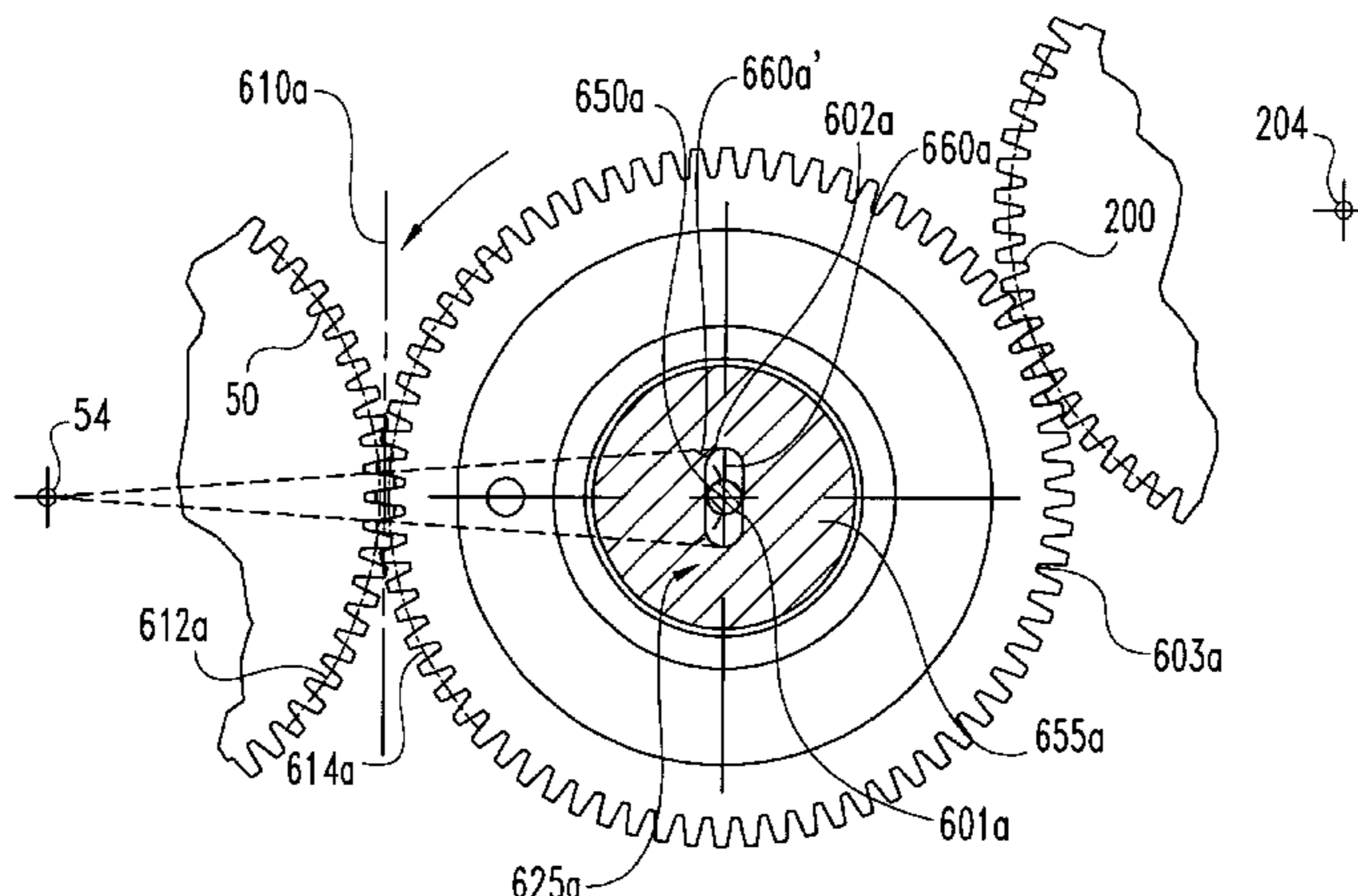
Apparatus and method for adjusting a gear in a gear train
assembly. A first gear with a first rotational center and a
second gear with a second rotational center are coupled to an
engine. A third gear with a third rotational center is provided
to form a first mesh with the first gear. The first mesh has a
predetermined minimum backlash. The third gear also forms
a second mesh with the second gear. The third gear includes
an adjustable positioning mechanism which allows the rota-
tional center of the third gear to be adjusted along a
predetermined adjustment path that is substantially tangent
to the first mesh. The adjustment mechanism of the third
gear allows the minimum backlash of the first mesh to be
maintained while the second mesh is adjusted to likewise
achieve a generally minimized backlash.

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10 Claims, 21 Drawing Sheets



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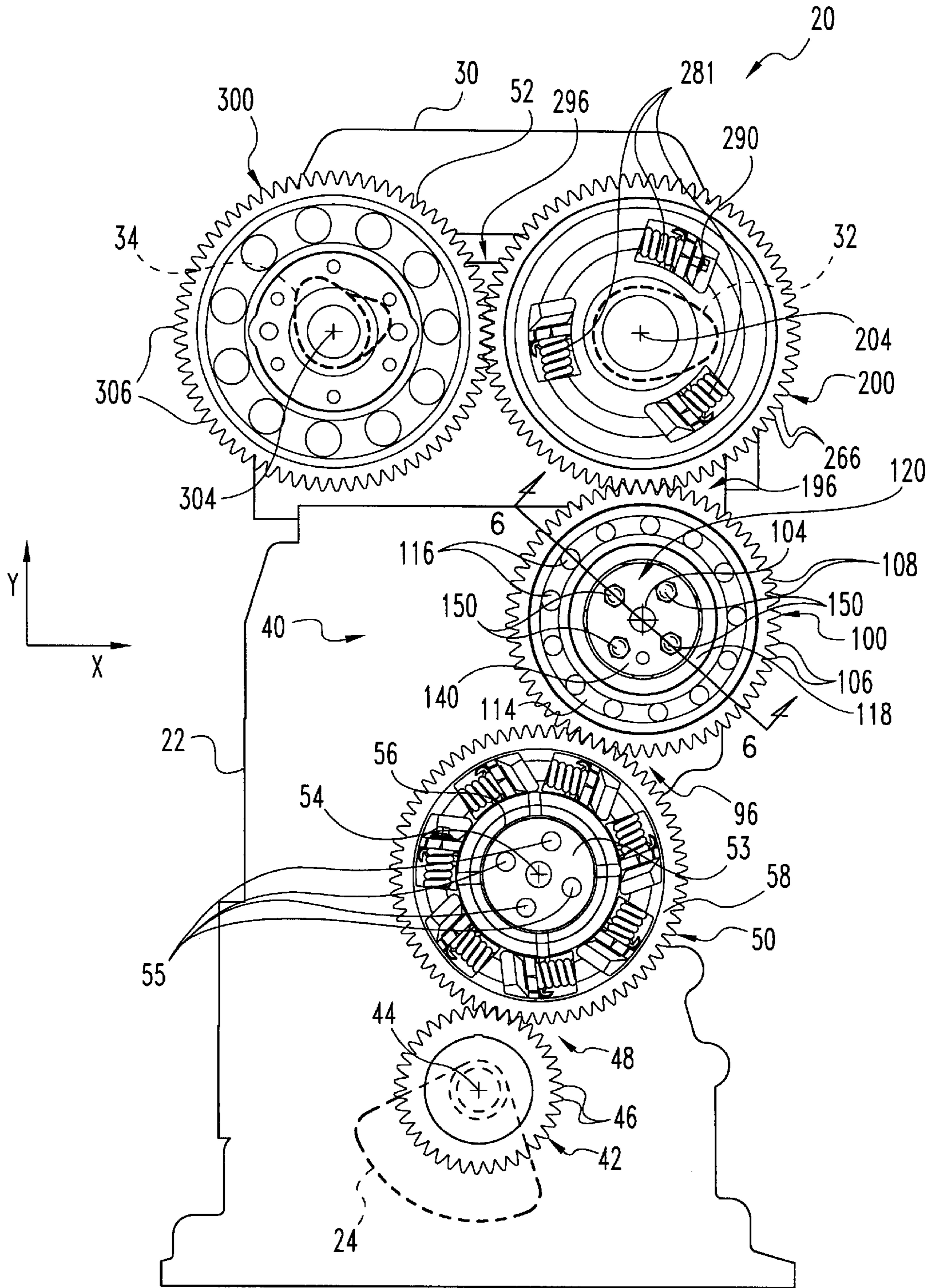


Fig. 1

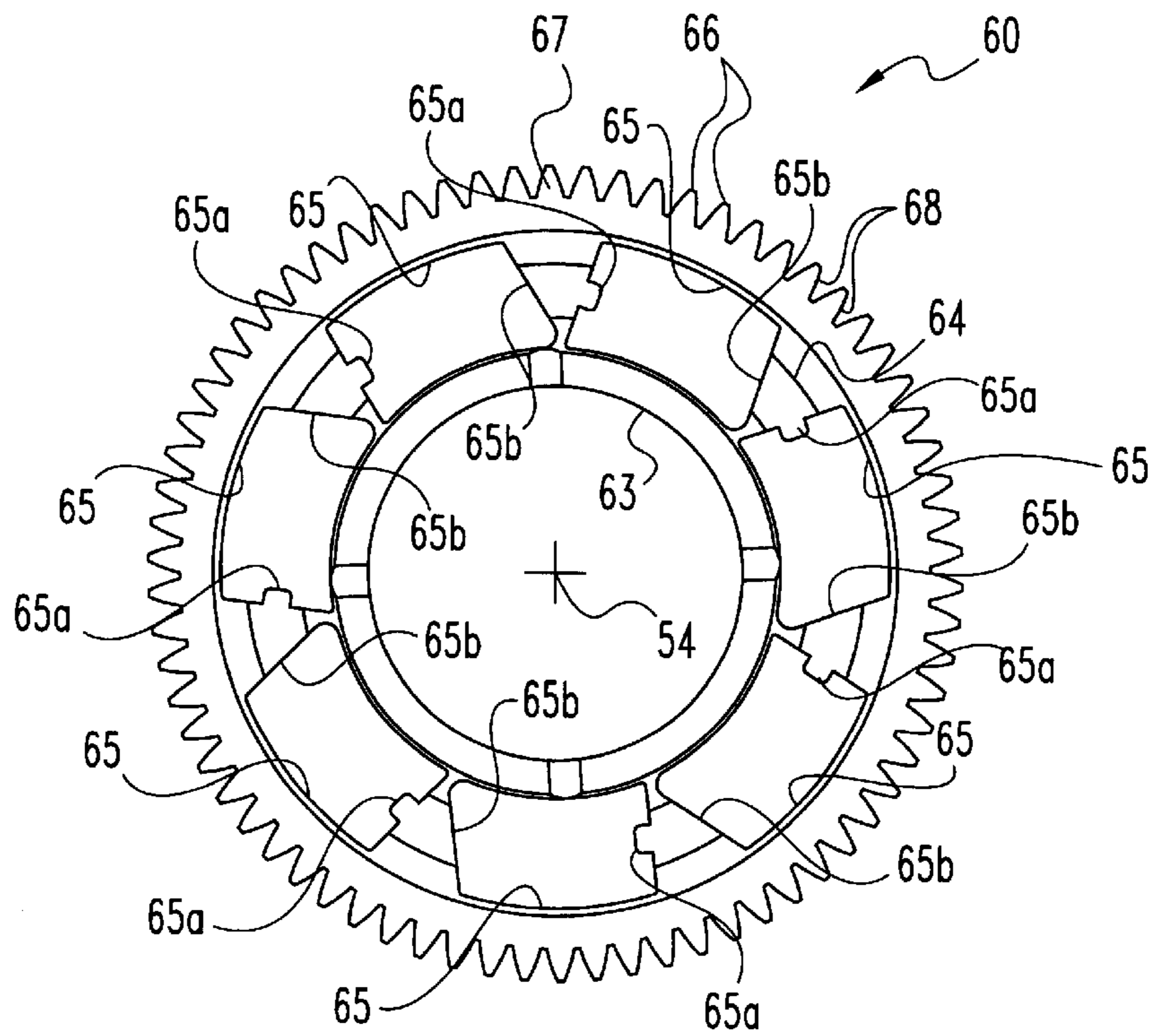


Fig. 2

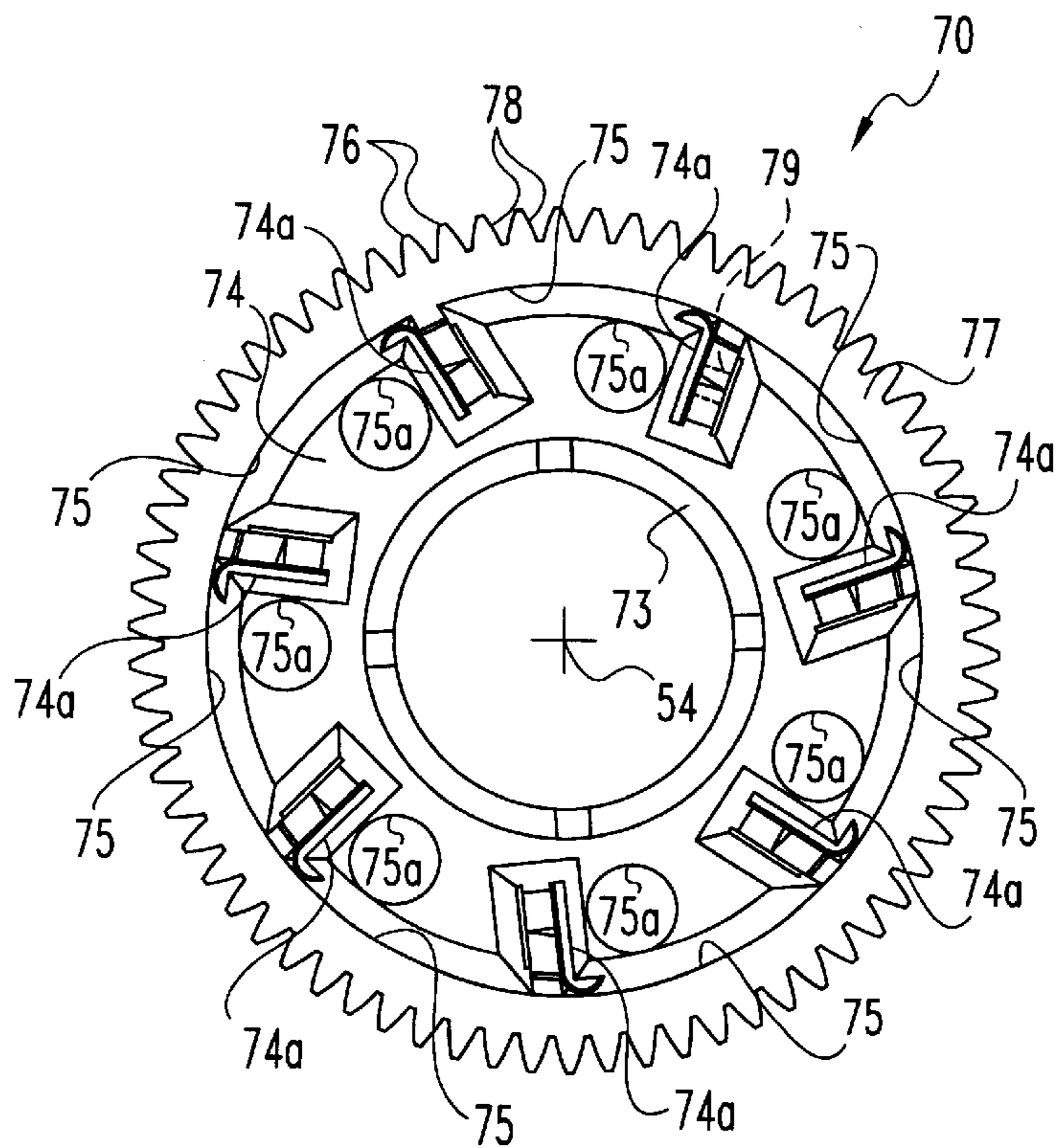


Fig. 3

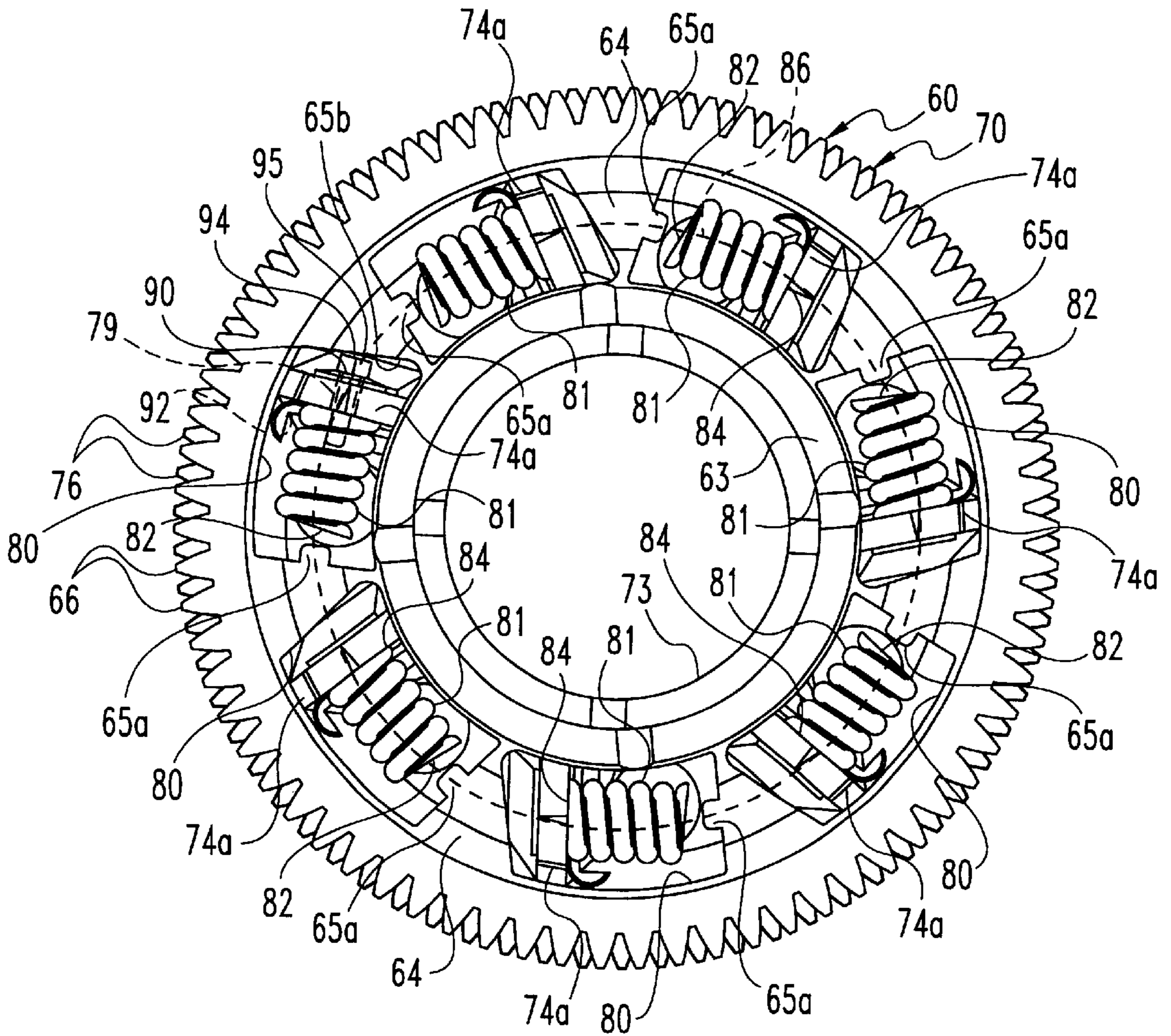


Fig. 4

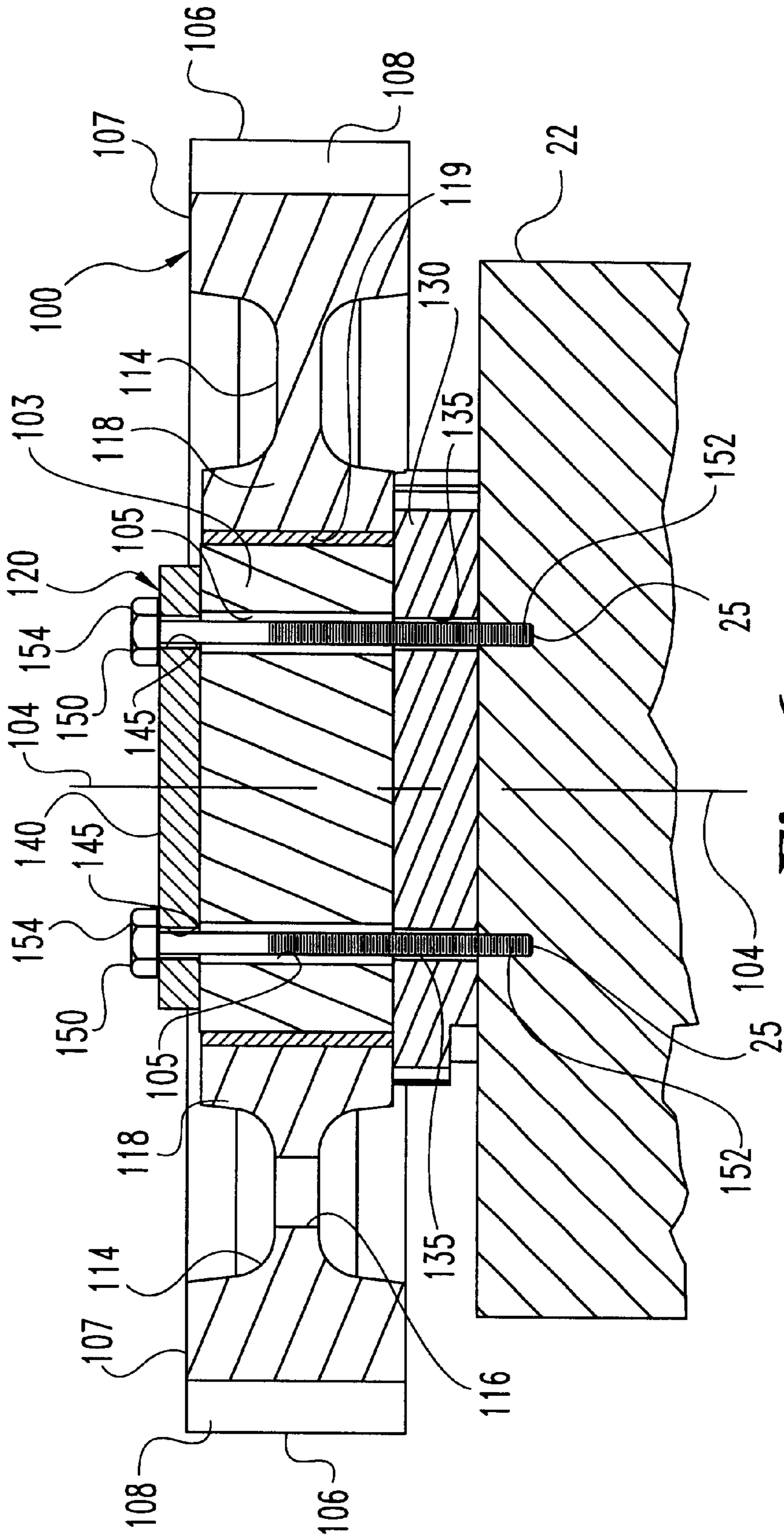


Fig. 6

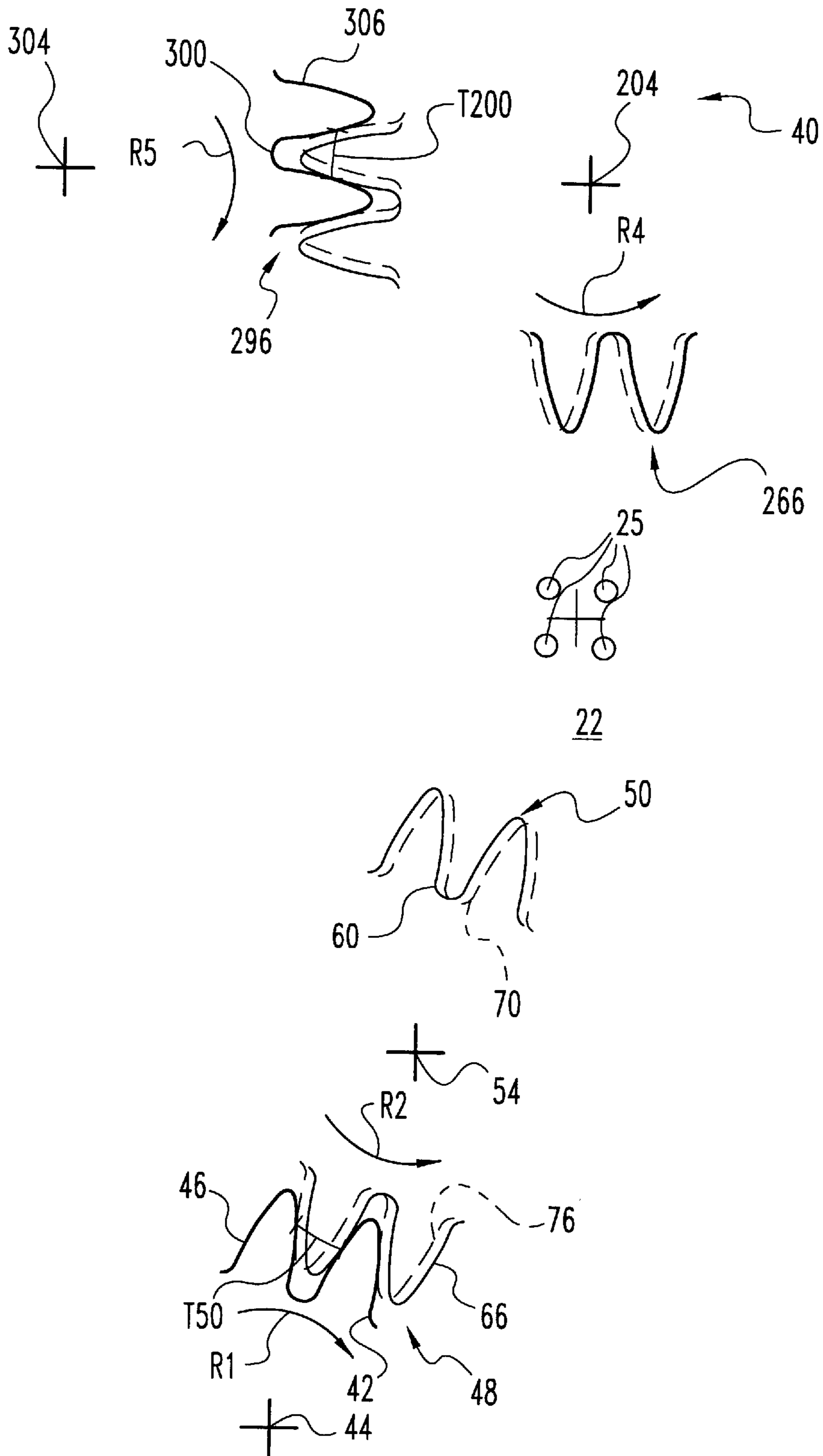


Fig. 7A

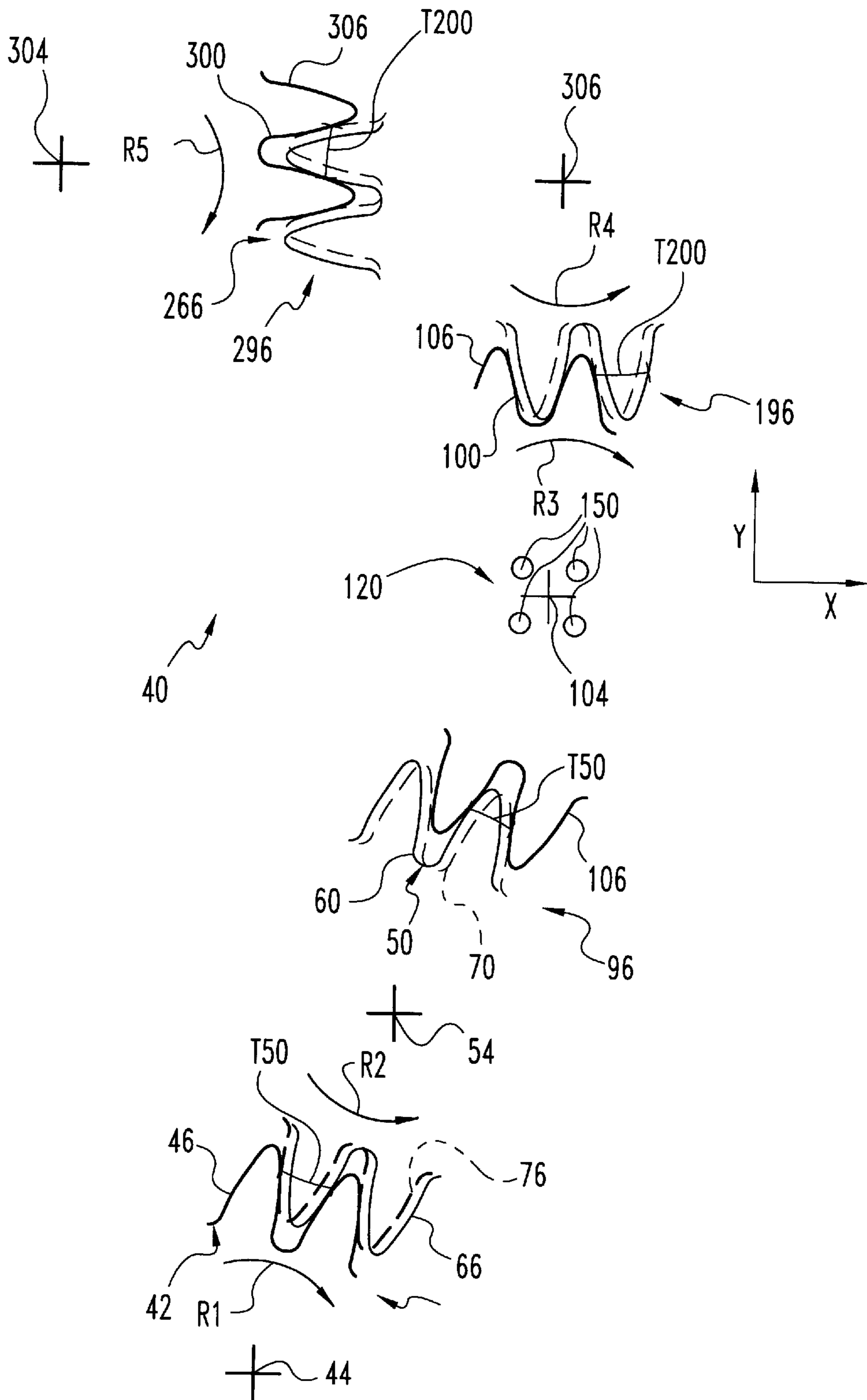


Fig. 7B

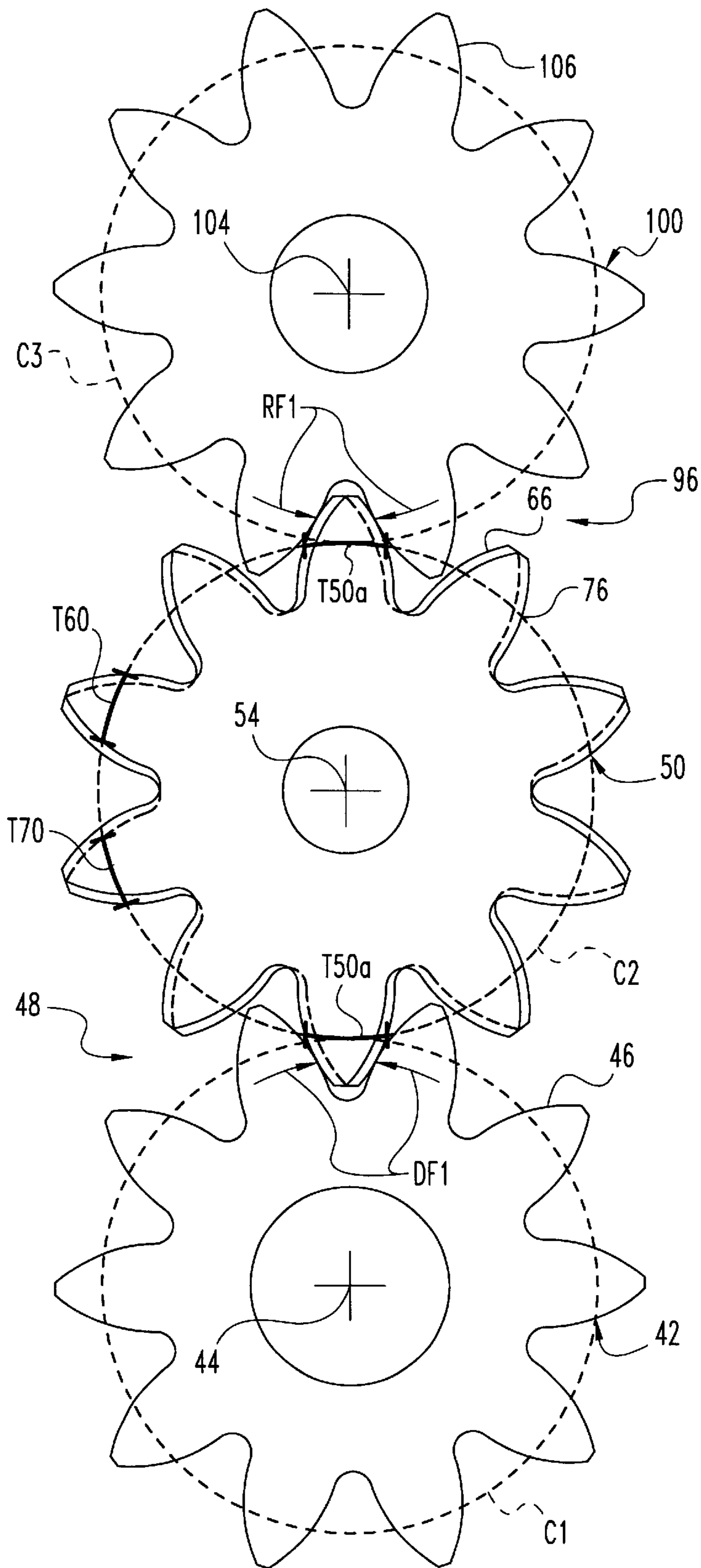


Fig. 8A

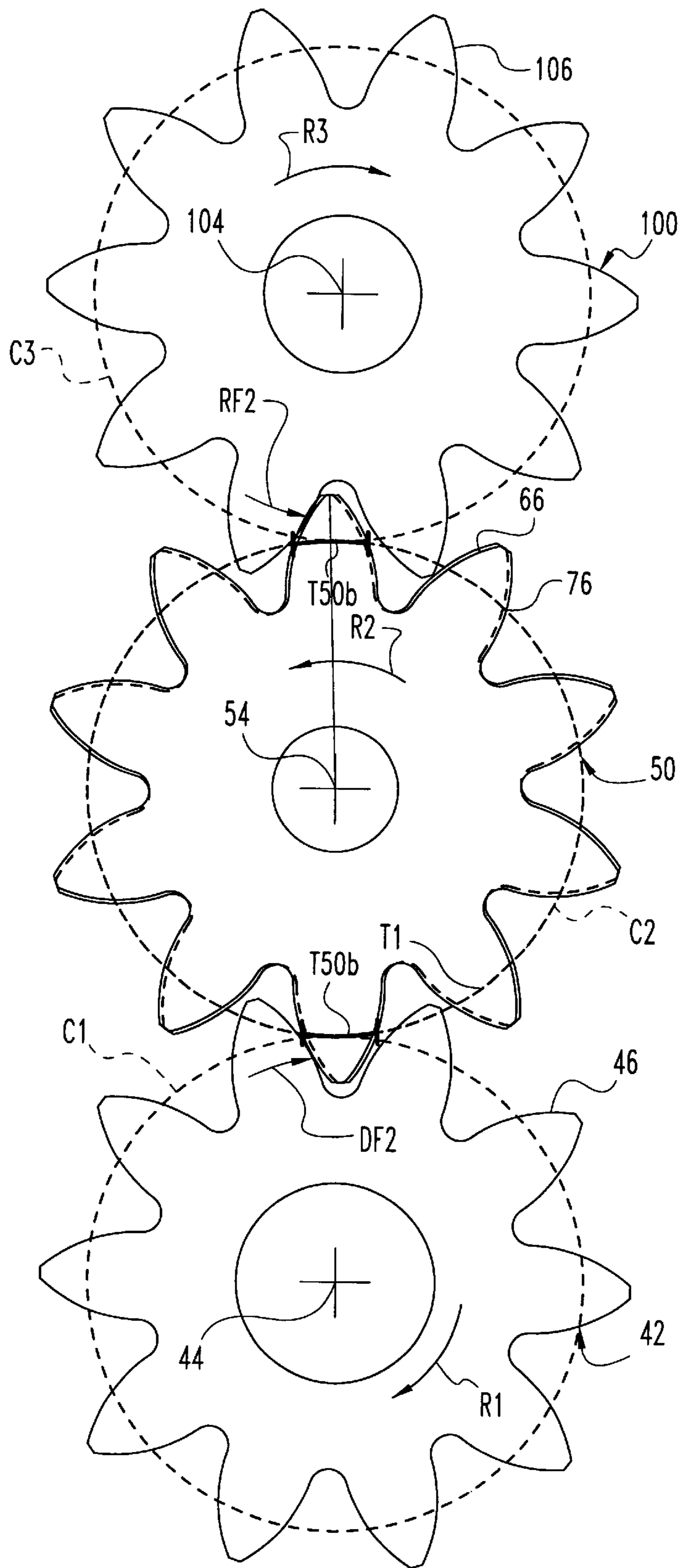


Fig. 8B

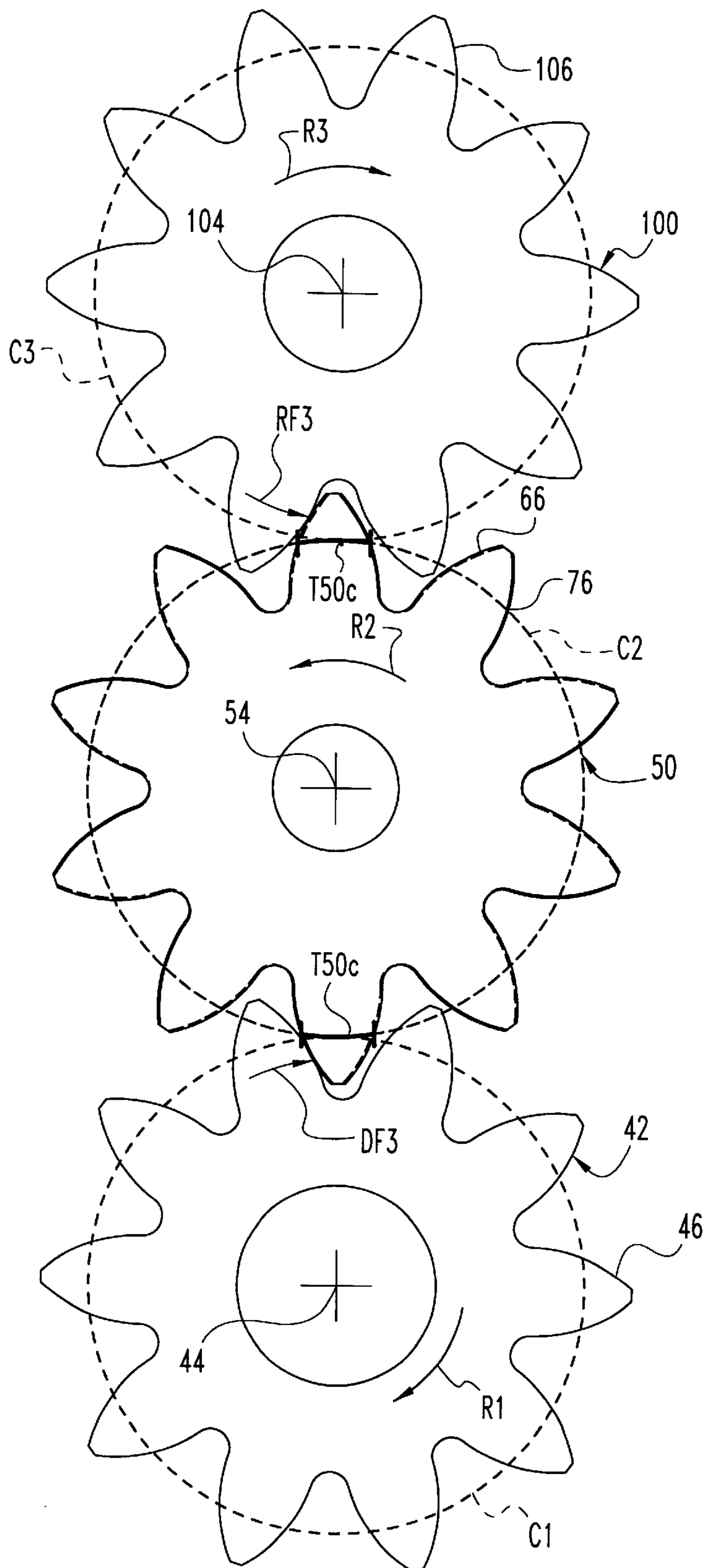


Fig. 8C

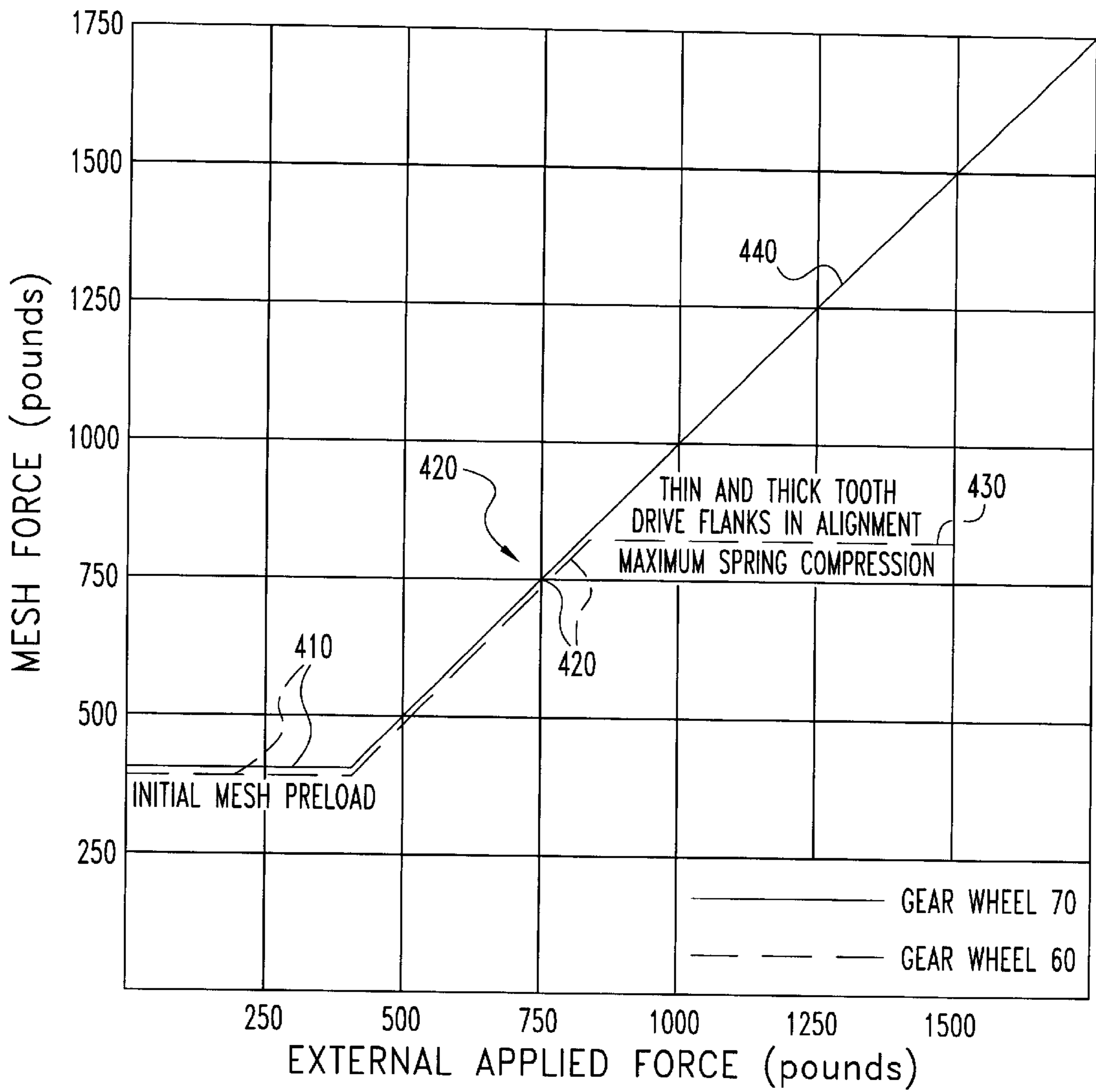


Fig. 9

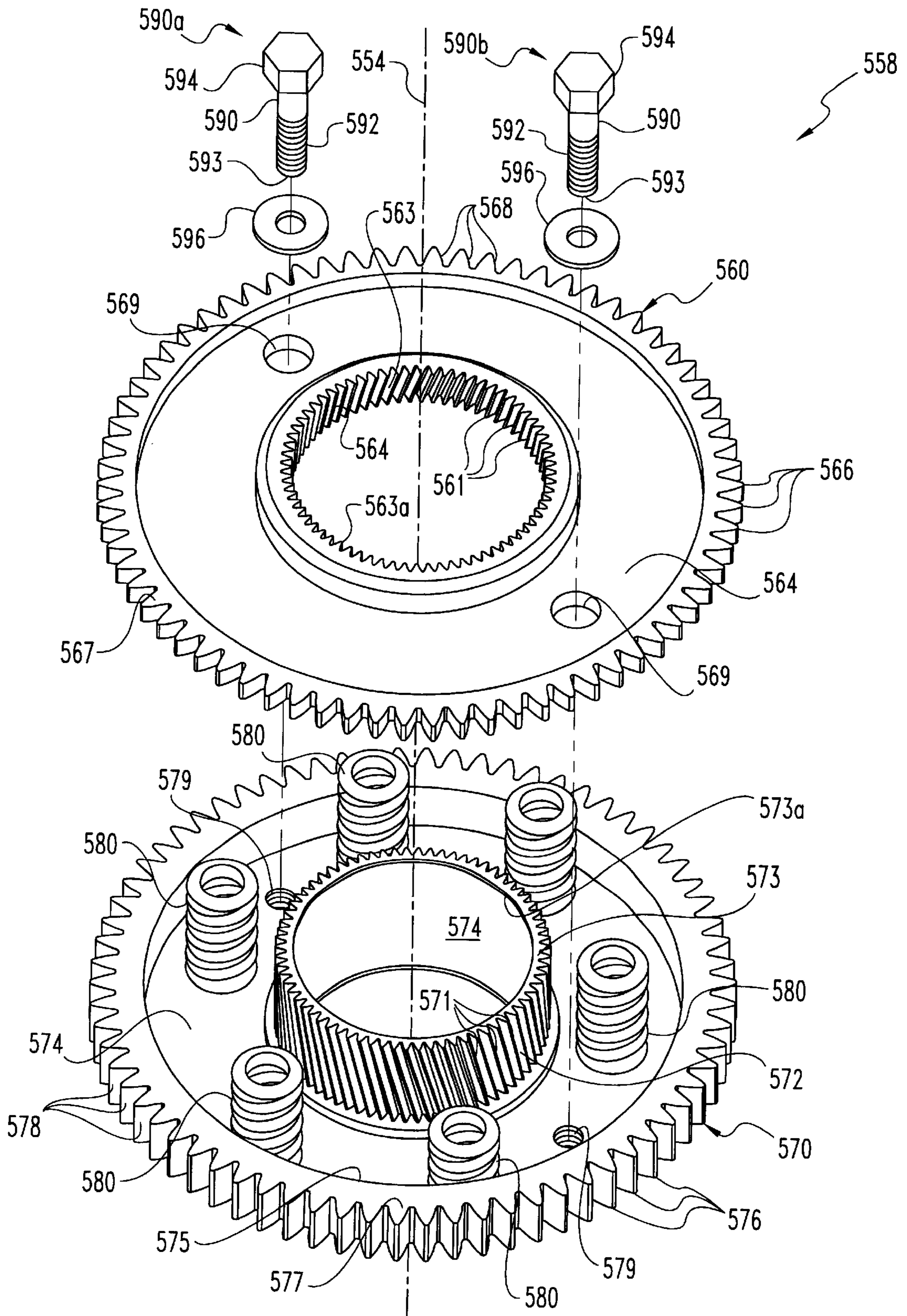


Fig. 10

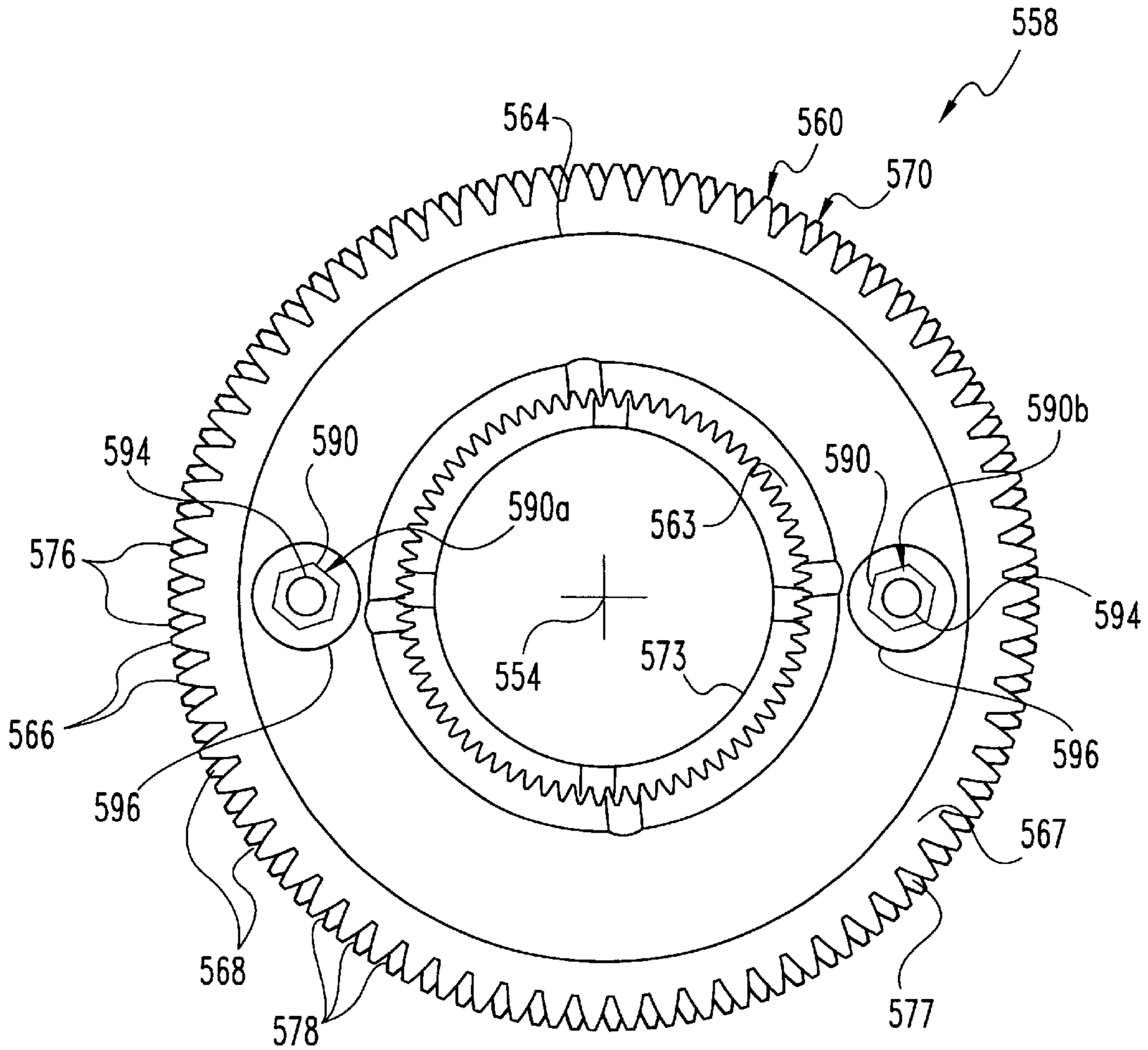


Fig. 11A

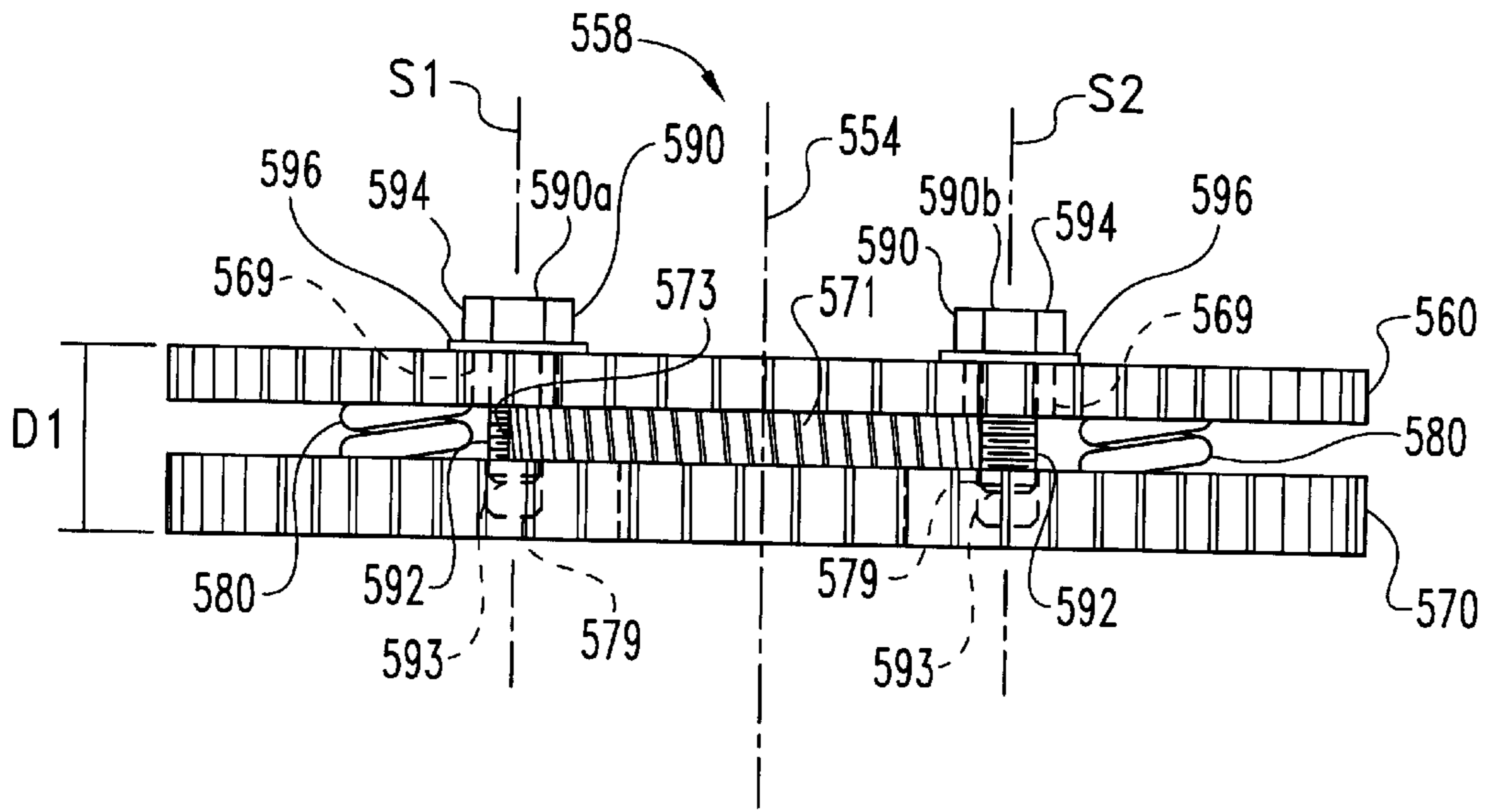


Fig. 11B

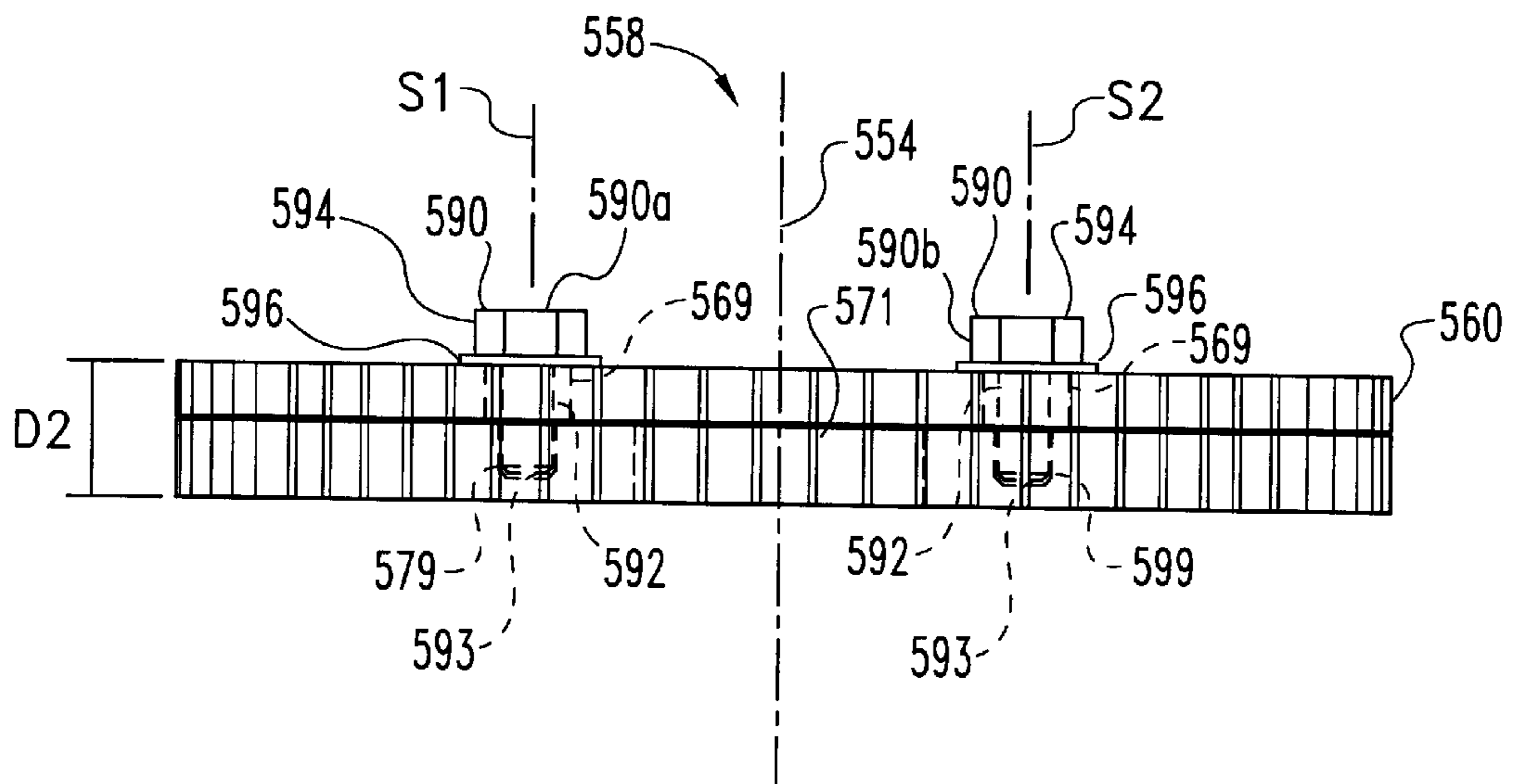


Fig. 12B

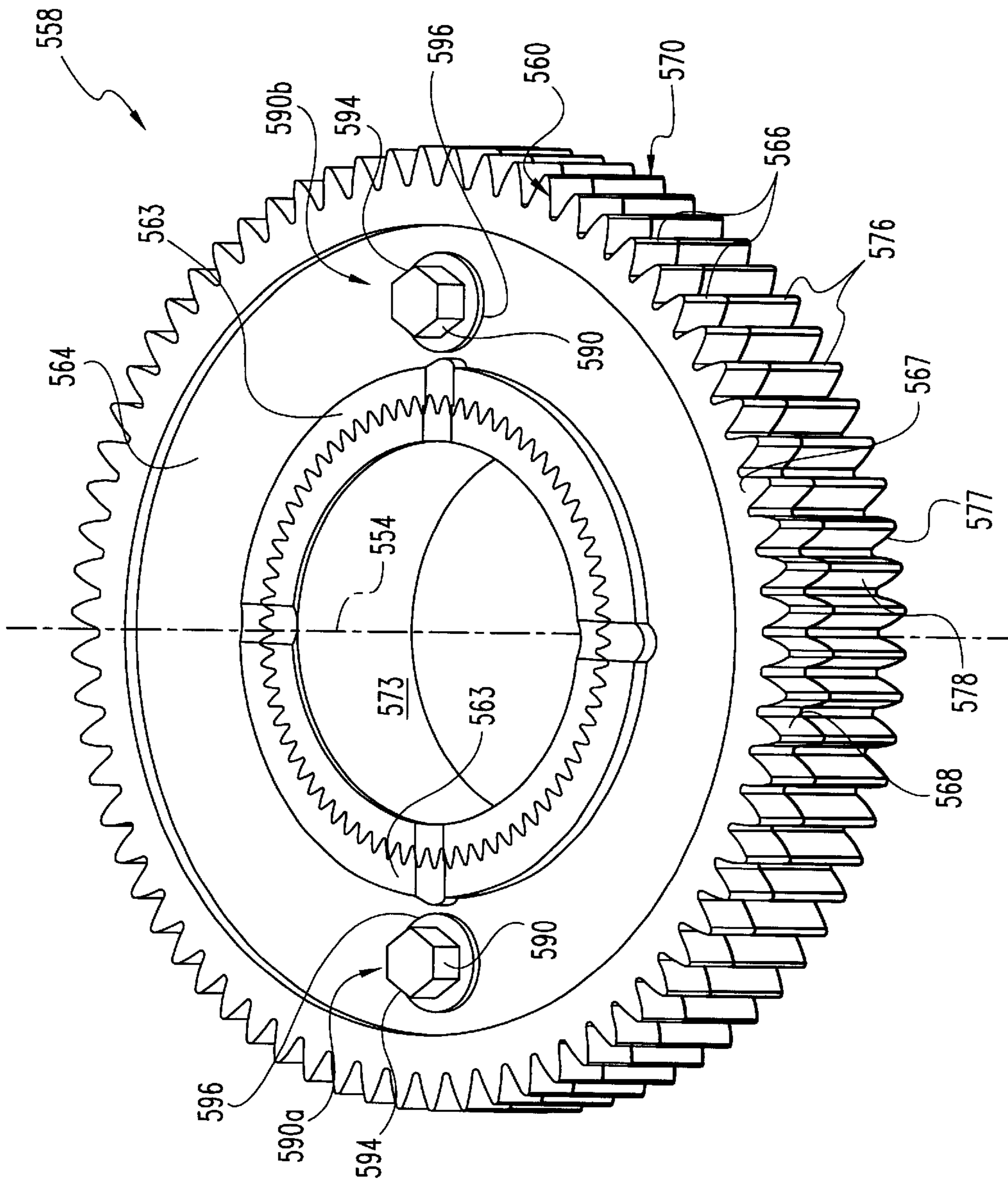


Fig. 12A

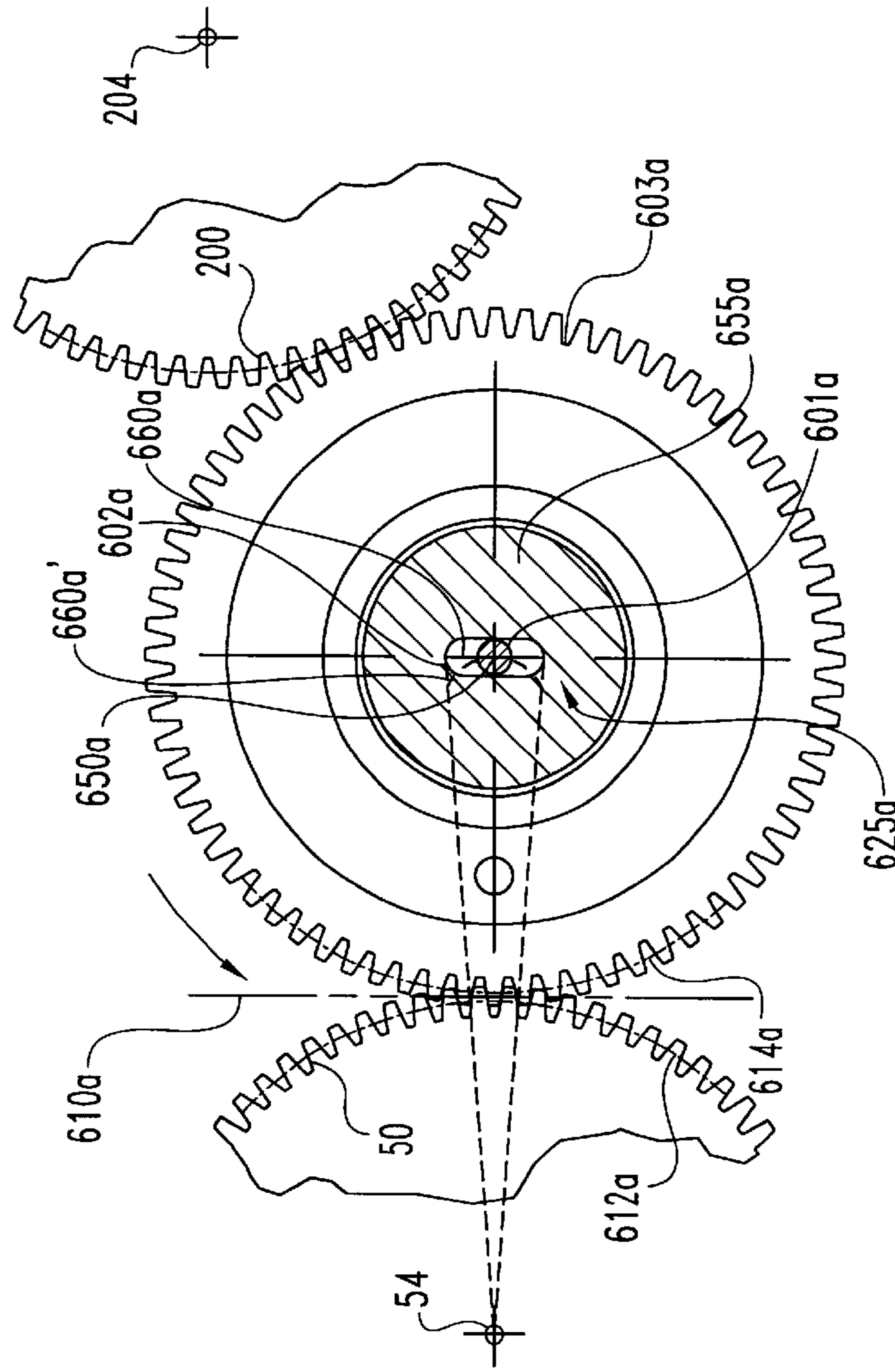


Fig. 14

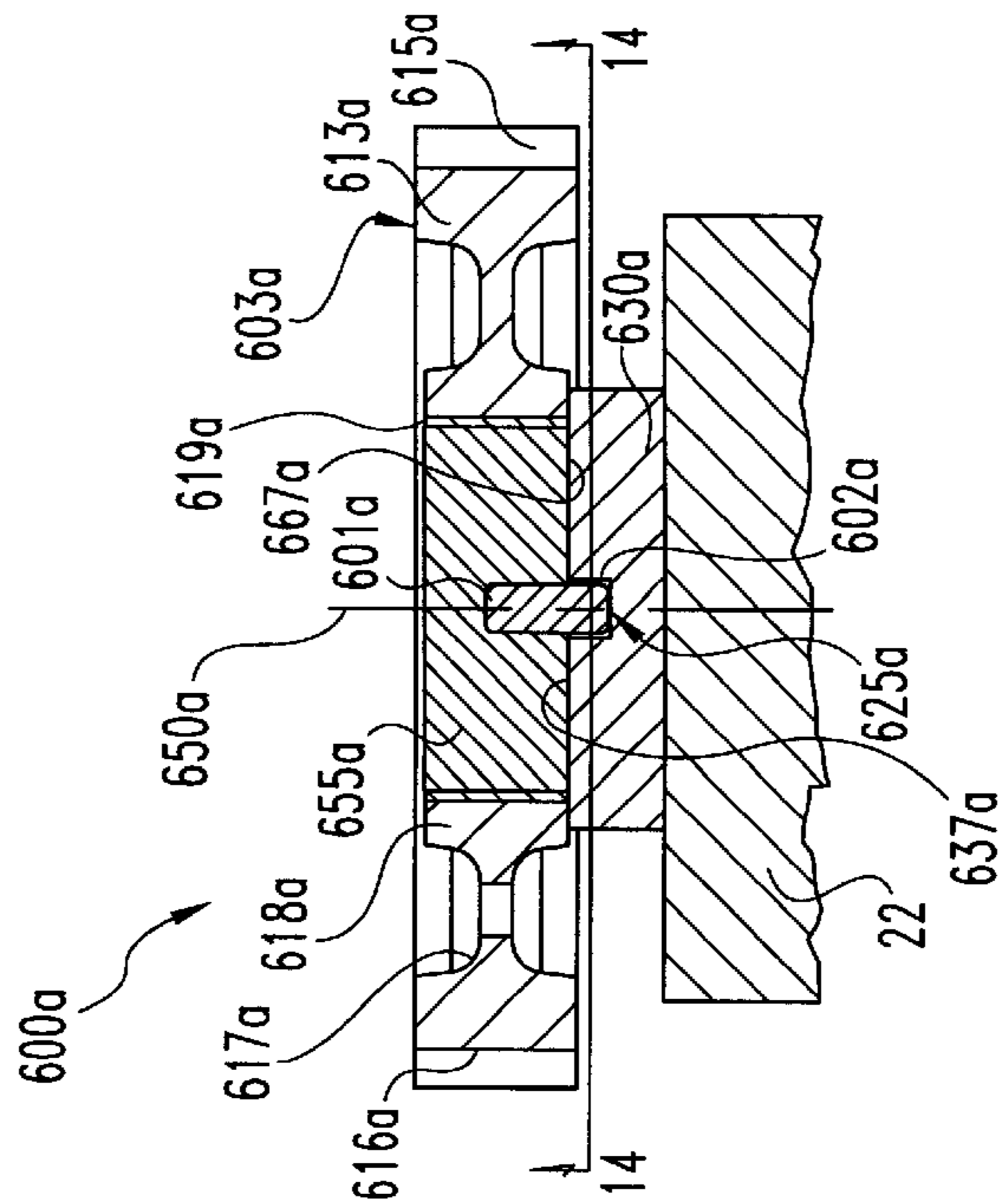


Fig. 13

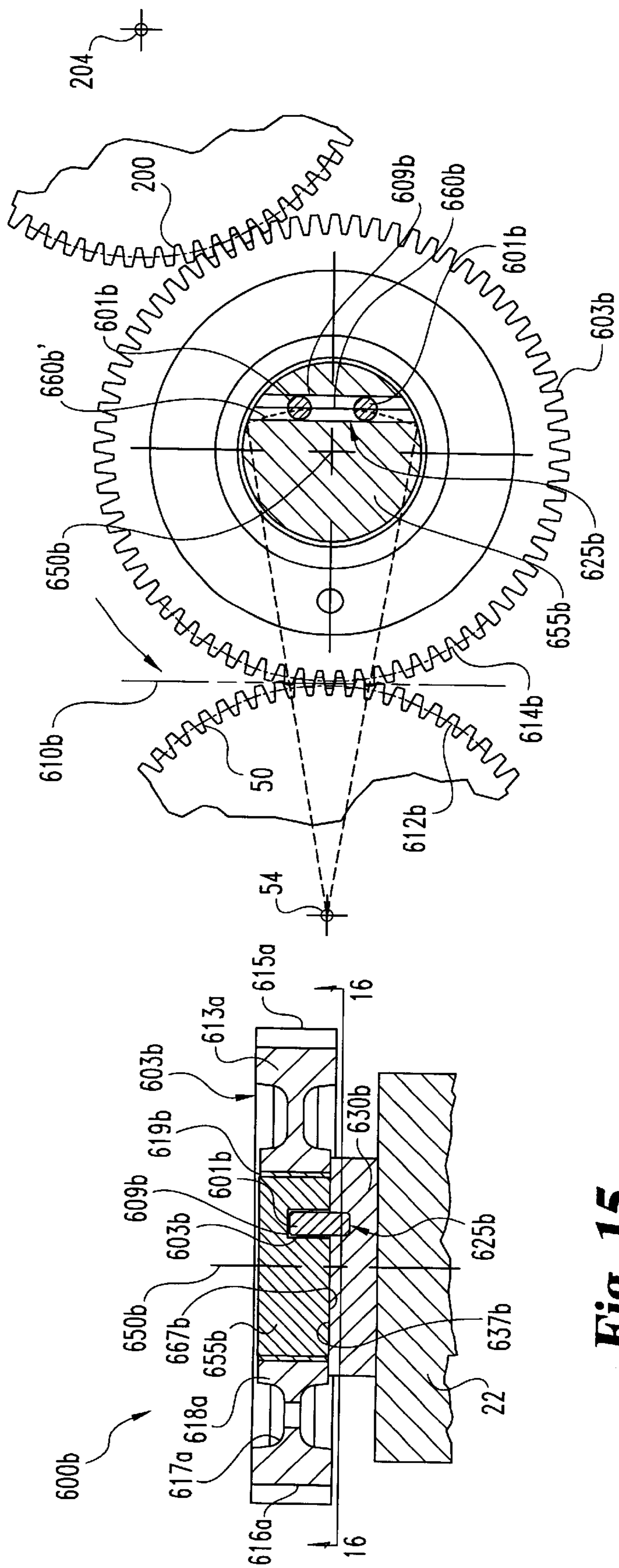


Fig. 16

Fig. 15

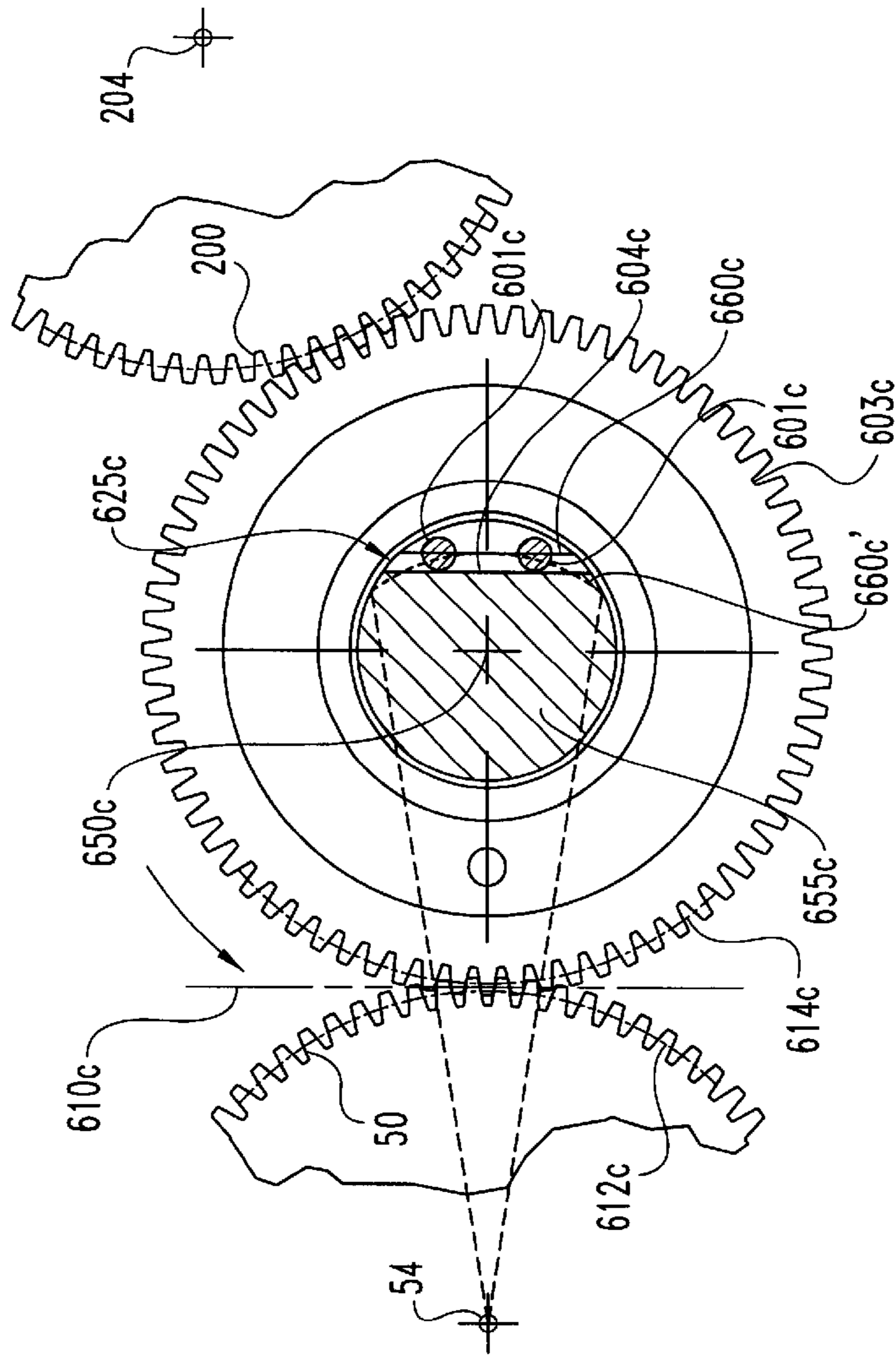


Fig. 17

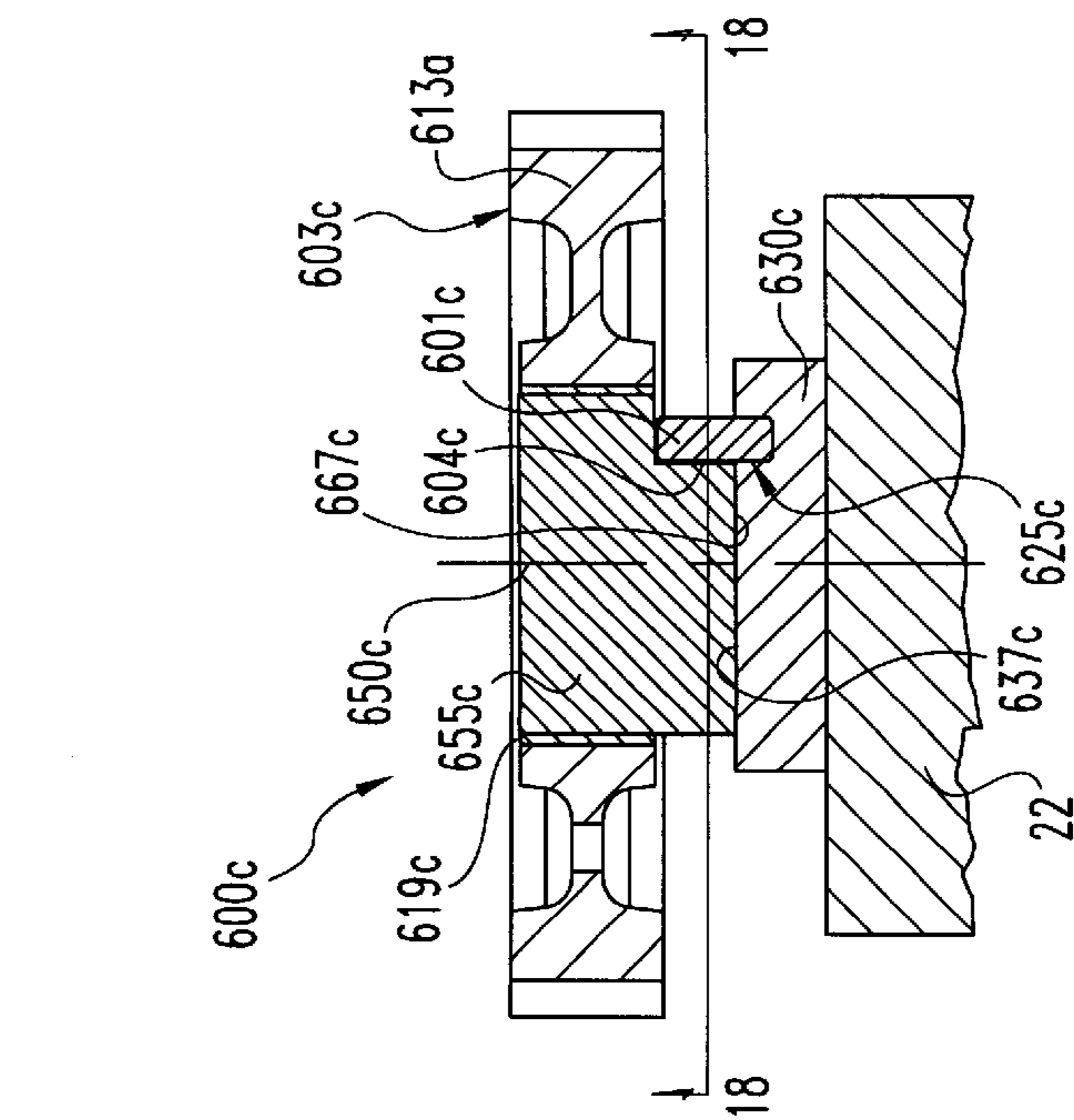


Fig. 18

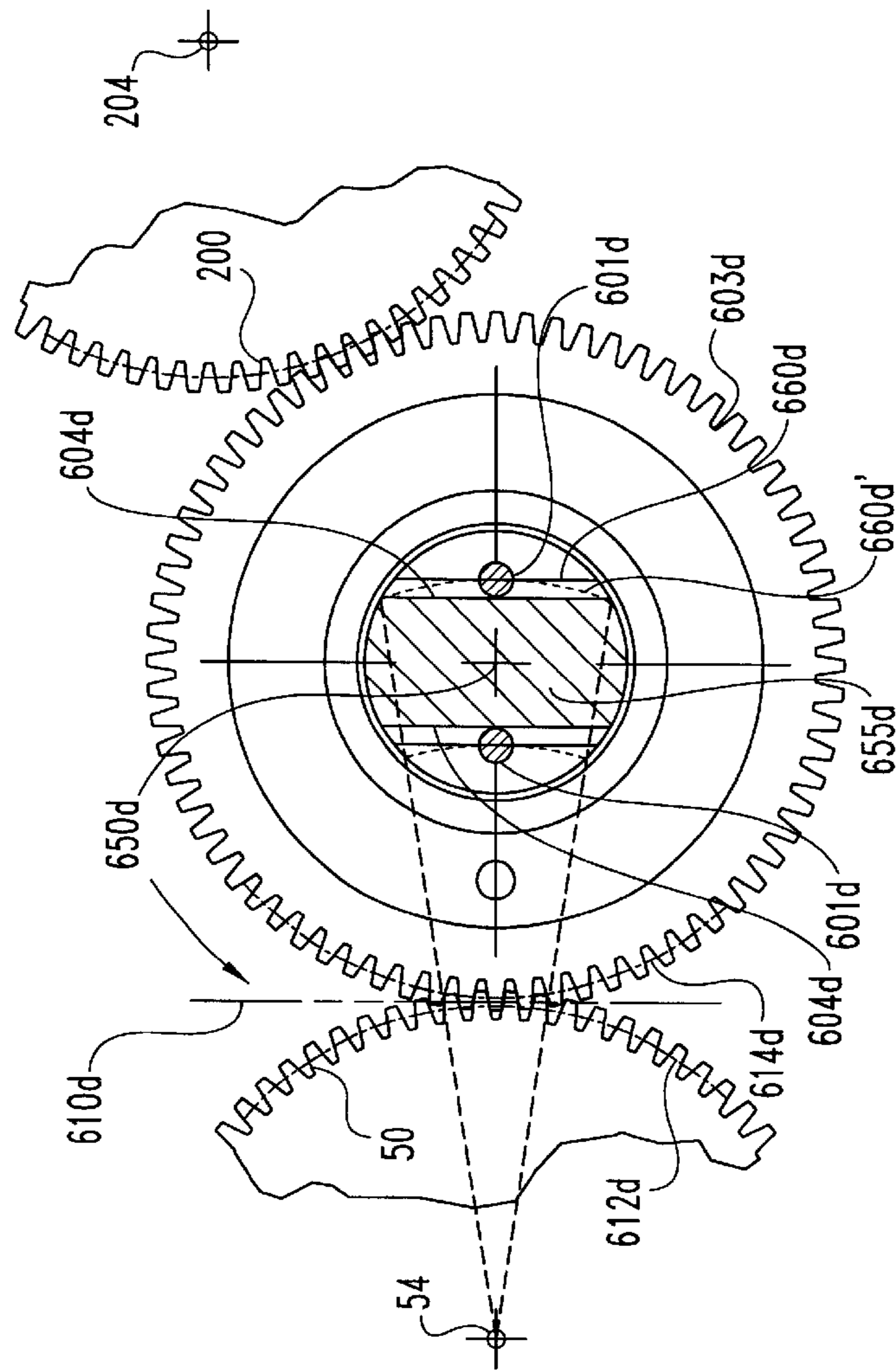


Fig. 20

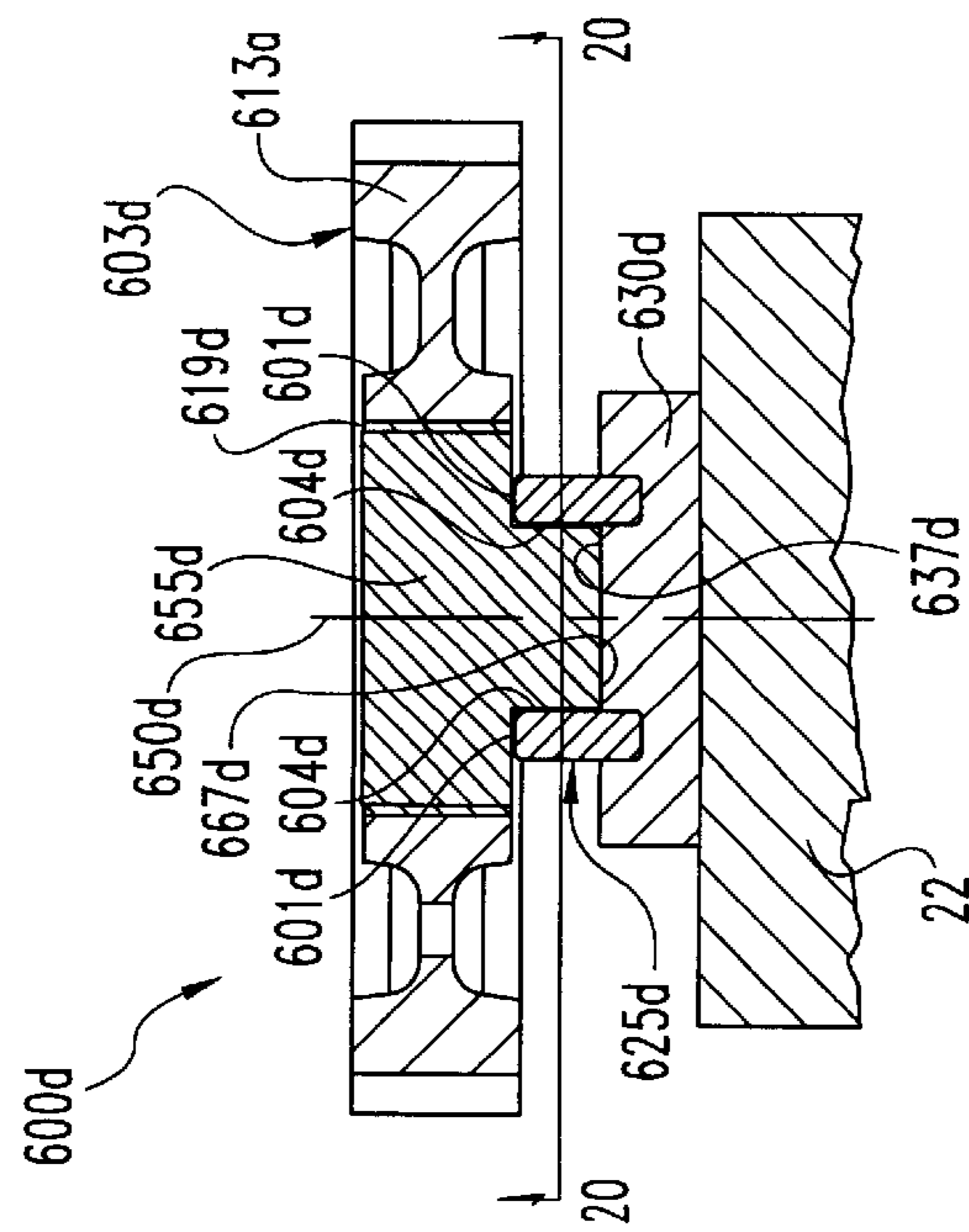


Fig. 19

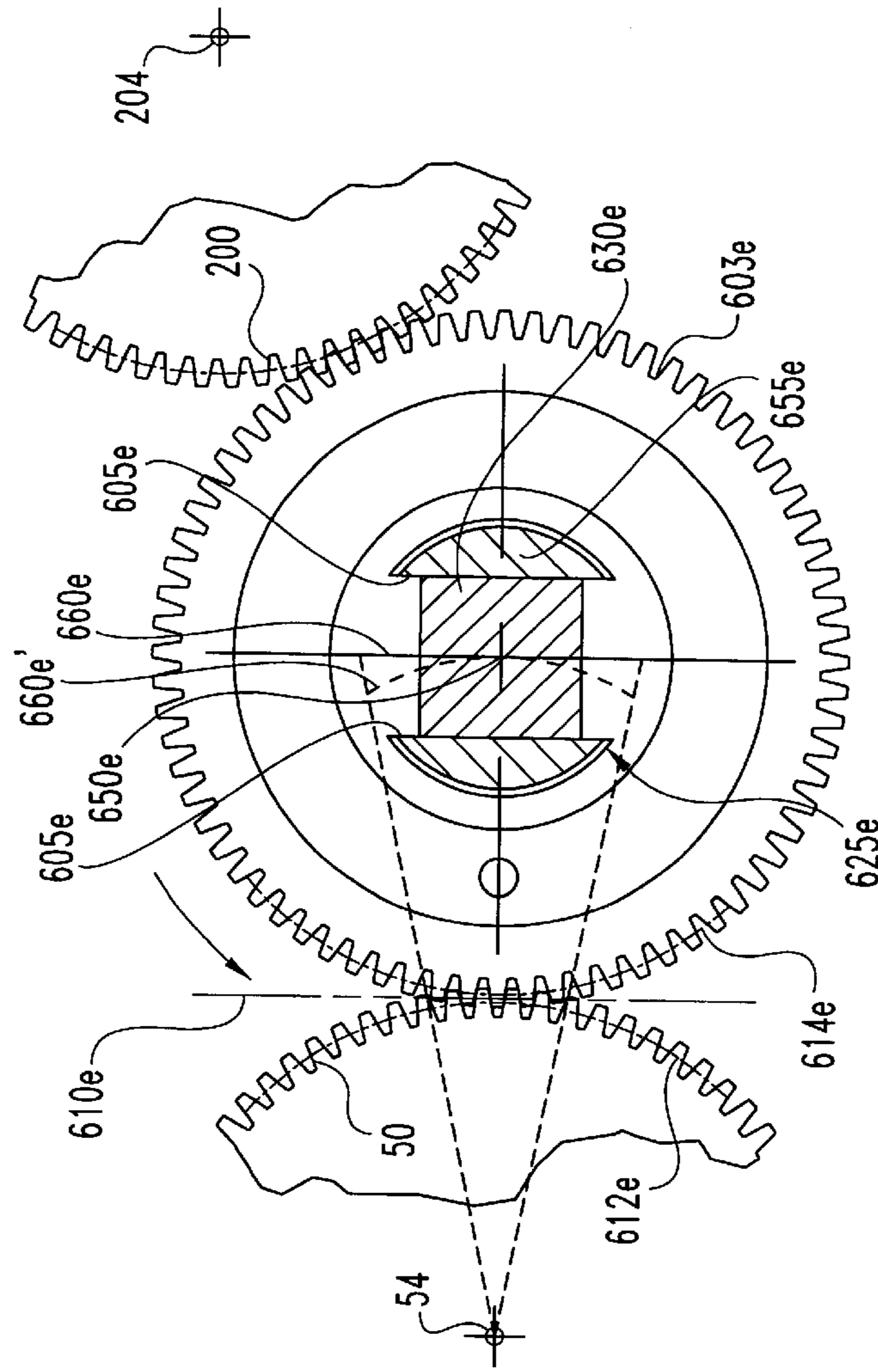


Fig. 22

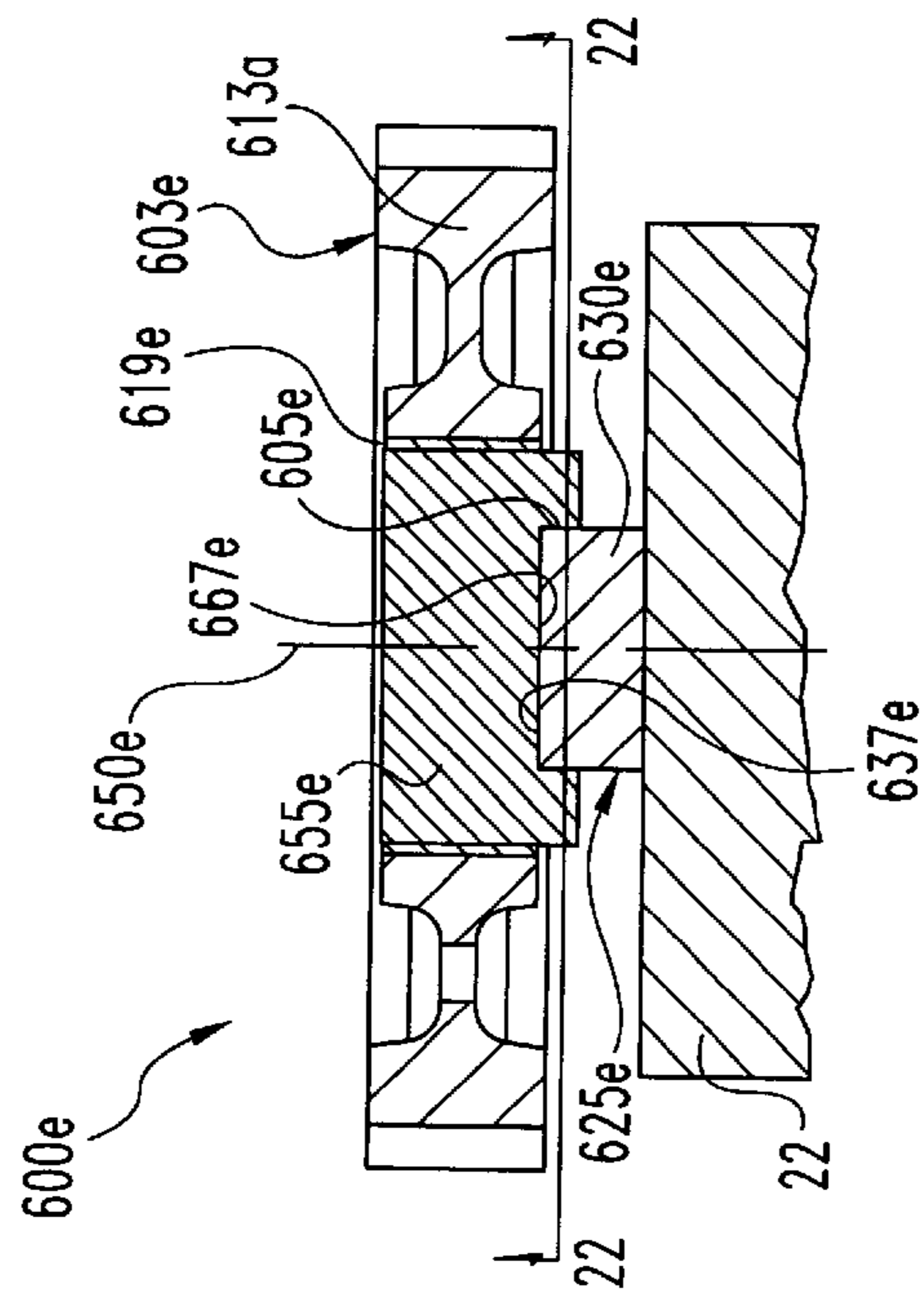


Fig. 21

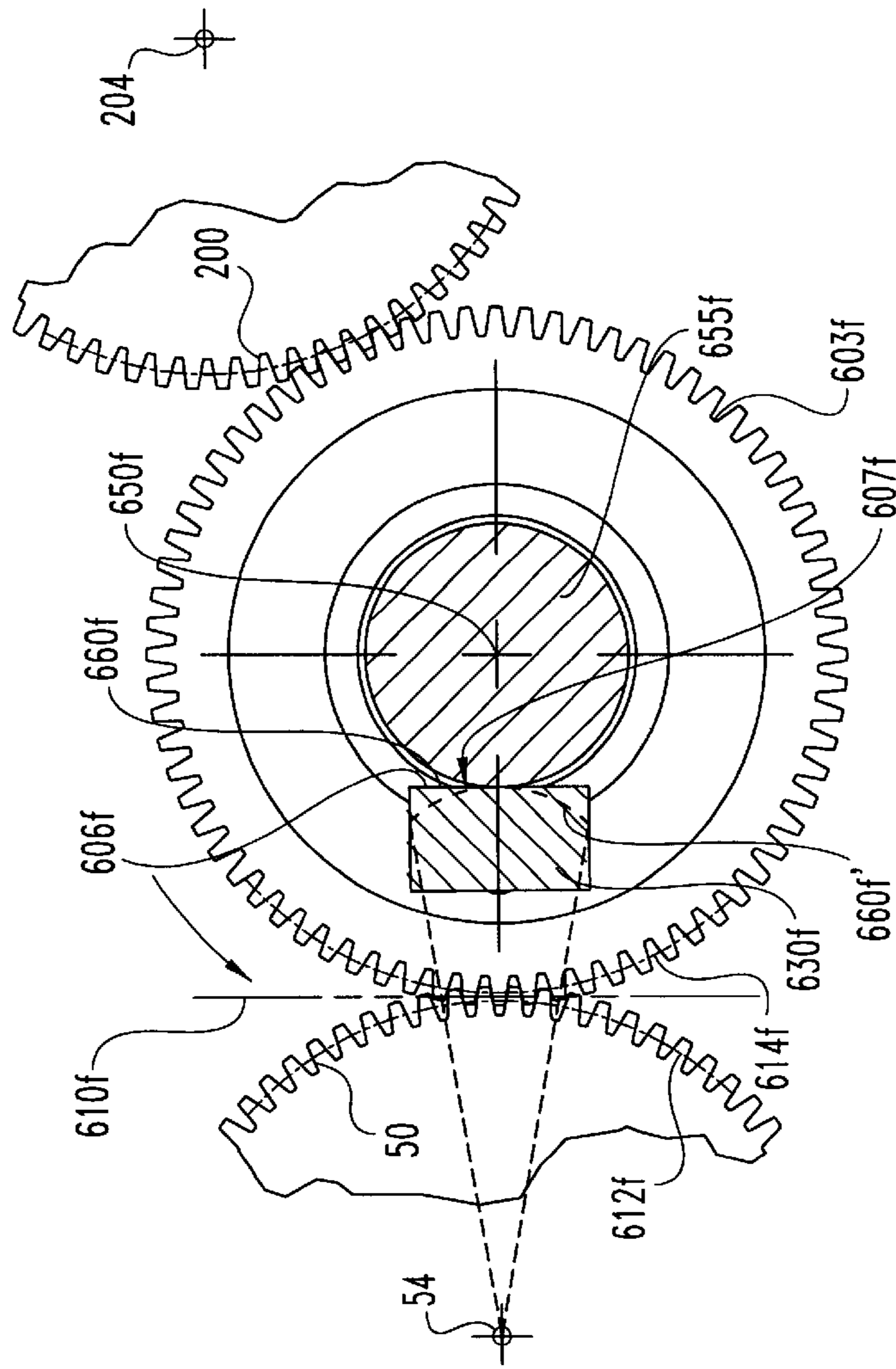


Fig. 24

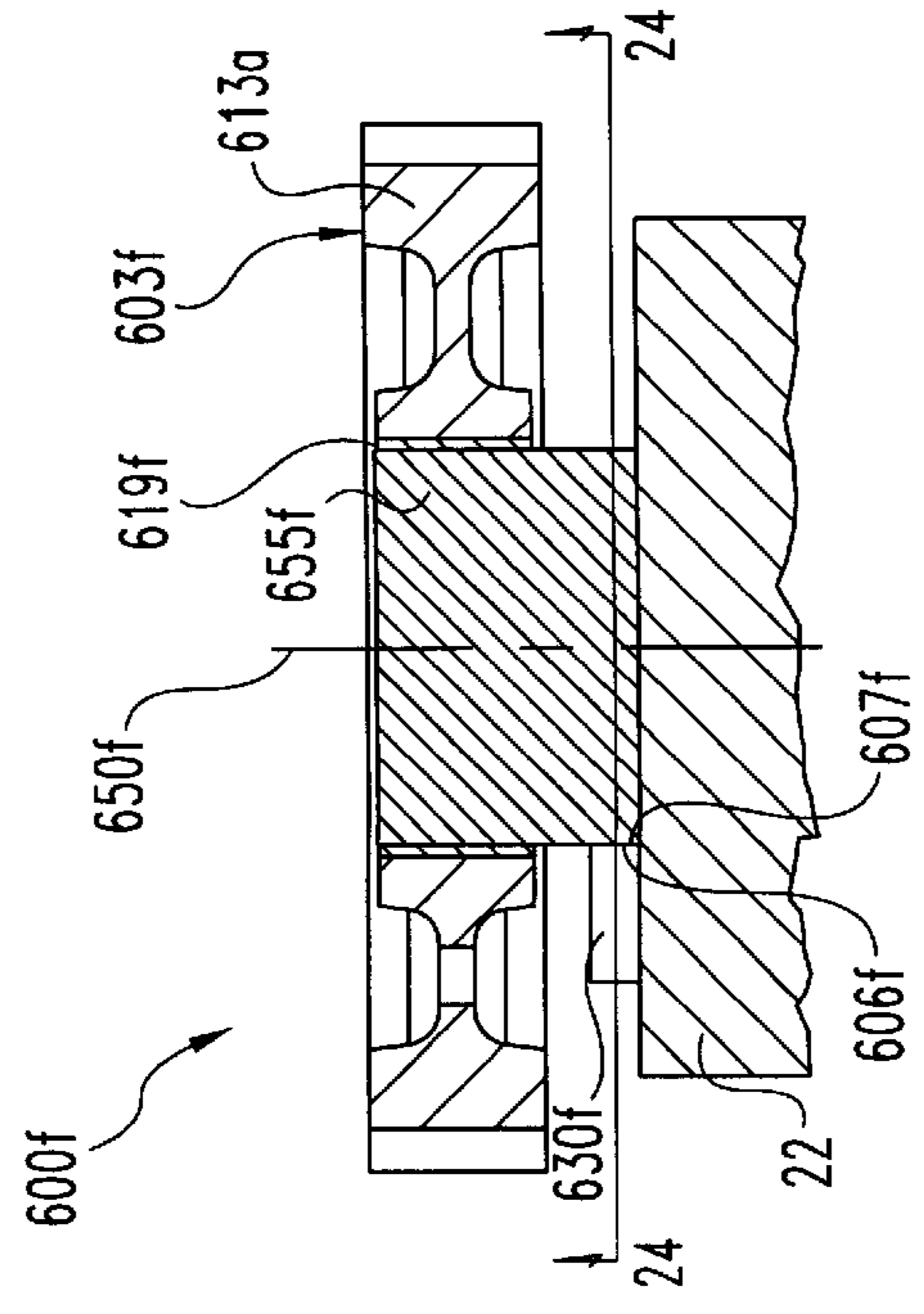


Fig. 23

APPARATUS AND METHOD FOR ADJUSTING A GEAR

REFERENCE TO RELATED APPLICATIONS

This application is a continuation of U.S. patent application Ser. No. 09/186,238 entitled APPARATUS AND METHOD FOR ADJUSTING A GEAR filed on Nov. 4, 1998, now U.S. Pat. No. 6,109,129, which is a continuation-in-part of U.S. patent application Ser. No. 08/853,341 entitled ANTI-LASH GEAR WITH ALIGNMENT DEVICE filed on May 8, 1997, now U.S. Pat. No. 5,870,928, and a continuation-in-part of U.S. patent application Ser. No. 08/853,013 entitled GEAR TRAIN ASSEMBLY INCLUDING SCISSOR GEAR filed on May 8, 1997, now U.S. Pat. No. 5,979,259. Additionally, this application is related to U.S. patent application Ser. No. 08/853,378 entitled ANTI-LASH GEAR ASSEMBLY filed on May 8, 1997, now U.S. Pat. No. 5,979,260.

BACKGROUND OF THE INVENTION

The present invention relates to gears, and more particularly, but not exclusively, relates to reduction of backlash in gear trains.

When the tooth of one gear mates with the gap of another gear, the gap typically provides more space than needed to accommodate the tooth. This excess space is sometimes called "lash" or "backlash." Backlash may vary with a number of factors including radial play in the gear bearings, gear shaft eccentricity, incorrect center-to-center spacing of the gears, and the gear-to-gear variation typical of many gear manufacturing processes.

The extra space associated with backlash usually leads to significant impact loading of the gear teeth. This loading often causes excessive noise and may result in other gear train problems. For example, backlash may accelerate gear wear. Backlash reduction is of particular concern for internal combustion engine applications—especially for gear trains used with diesel engines. U.S. Pat. No. 5,450,112 to Baker et al., U.S. Pat. No. 4,920,828 to Kameda et al., U.S. Pat. No. 4,700,582 to Bessette, and U.S. Pat. No. 3,523,003 to Hambric are cited as sources of background information concerning the application of gear trains to various engines.

One way to reduce backlash is through precision machining and mounting of the gears. However, this approach is usually expensive and still may not adequately address backlash that changes over time due to wear. Another approach to reduce backlash has been the introduction of one or more scissor gears into the gear train. Generally, scissor gears have teeth which adjust in size to occupy the space available between teeth of a mating gear. U.S. Patent No. 5,056,613 to Porter et al., U.S. Pat. No. 4,747,321 to Hannel, U.S. Pat. No. 4,739,670 to Tomita et al., U.S. Pat. No. 3,365,973 to Henden, and U.S. Pat. No. 2,607,238 English et al. are cited as examples of various types of scissor gears.

Backlash accommodation with a scissor gear is often limited when the scissor gear is meshed with two or more gears having different amounts of lash. Typically, the mating gear having the smallest amount of lash dictates the effective tooth size of the scissor gear; however, this size is generally inadequate to take-up the greater lash of the other mating gear or gears. One potential solution to this problem is to select mating gears which minimize the lash difference, but this "lash matching" approach is typically expensive and time-consuming. Consequently, a need remains for a gear train assembly which accommodates lash differences resulting from multiple gears meshing with a scissor gear.

One scissor gear configuration has two toothed wheels spring-biased to rotate relative to each other about a common center. For this configuration, paired gear teeth, one from each wheel, spread to occupy the available space between teeth in a mating gear. In some gear trains, loading of the tooth pairs by the mating gear becomes high enough to align each tooth pair in opposition to the spring bias. Typically, each member of the aligned pair is configured to proportionally bear this high load by being sized with the same nominal thickness. However, it has been found that random deviations from nominal are usually enough to cause one tooth or the other of each pair to bear a disproportionately high amount of the load until it has deformed enough to match the other tooth. This deformation process often subjects the gear teeth to reverse bending loads that more quickly wear-out the teeth compared to teeth subjected to unidirectional bending loads. Also, such deformation may cause greater tooth-to-tooth variation, resulting in poorer performance and a more noisy gear train. Therefore, a need exists for an anti-lash gear assembly which accommodates high loading without these drawbacks.

It has also been discovered that the knocking of heavy duty diesel engines, often attributed to combustion processes, results, at least in part, from high impact gear tooth noise. Typically, this noise is not sufficiently abated by conventional scissor gear configurations. Thus, a gear train is also in demand which addresses this type of noise.

SUMMARY OF THE INVENTION

The present invention relates to anti-lash gear assemblies and gear trains utilizing one or more anti-lash gear assemblies. Various aspects of the invention are novel, nonobvious, and provide various advantages. While the actual nature of the invention covered herein can only be determined with reference to the claims appended hereto, certain forms of the invention that are characteristic of the preferred embodiments disclosed herein are described briefly as follows.

One form of the present invention is a gear train having a first gear forming a first mesh with a second gear and a second mesh with a third scissor gear. A mounting position of the first gear relative to the second and third gears is selected to maintain backlash of both the first and second meshes at or below a maximum acceptable level.

In an another form, an idler gear forms a first mesh with a first scissor gear to establish an effective tooth size of the first scissor gear. After the first mesh is established, the mounting position of the idler gear is selected. This mounting position is determined as a function of the effective tooth size to control backlash of a second mesh formed between the idler gear and a second scissor gear.

In a further form, an idler gear forms a first mesh with a first gear and a second mesh with a second gear. A mounting position of the idler gear relative to the first and second gears is selected to maintain backlash of both the first and second meshes at or below a maximum acceptable level. This mounting position may be selected by using a mechanism that constrains movement of the idler gear along a desired path and the first gear, second gear, or both may be of a scissor type configuration.

In an additional form of the present invention, a gear train assembly for an engine is provided which includes a first scissor gear rotatably coupled to the engine, the first scissor gear having a first rotational center; a second scissor gear rotatably coupled to the engine, the second scissor gear having a second rotational center; and an idler gear rotatably

coupled to the engine to form a first mesh with the first scissor gear and a second mesh with the second scissor gear. The idler gear has a third rotational center and is mounted by a positioning mechanism having a guide member slidingly engaging a generally linear adjustment path, the idler gear being selectively positionable along the path to maintain a distance between the first and third rotational centers within a predetermined range corresponding to a generally minimized backlash for the first mesh, and to provide a correspondingly adjustable separation distance range between the second and third rotational centers to generally match backlash of the second mesh to the minimized backlash of the first mesh.

In yet another form of the present invention, a method of assembling a gear train includes: (a) mounting a first gear which defines a first rotational center; (b) mounting a second gear which defines a second rotational center; and (c) mounting a third gear defining a third rotational center to establish a first mesh of a generally minimized backlash with the first gear and a second mesh with the second gear, the third gear being adjustable about a generally linear adjustment path, the linear adjustment path having a predetermined relationship to a tangent formed by the first mesh, whereby the third gear is selectively positionable along the adjustment path to minimize a backlash of the second mesh while maintaining the generally minimized backlash of the first mesh.

In still another form of the present invention, a gear train assembly for an engine includes a first gear rotatably coupled to the engine, the first gear having a first rotational center; a second gear rotatably coupled to the engine, the second gear having a second rotational center; and an idler gear rotatably coupled to the engine to form a first mesh with the first gear and a second mesh with the second gear. The idler gear defines a third rotational center and includes means for positioning the third gear along a generally linear adjustment path, the idler gear being selectively positionable along the path to maintain a distance between the first and third rotational centers within a predetermined range corresponding to a generally minimized backlash for the first mesh and to provide a correspondingly adjustable separation distance range between the second and third rotational centers to generally match backlash of the second mesh to the minimized backlash of said first mesh.

Other forms of the present invention include incorporating the various anti-lash gear assemblies of the present invention into a gear train and utilizing the adjustable positioning mechanisms of the present invention with an internal combustion engine.

It one object of the present invention to reduce noise emitted by engine gear trains.

Another object of the present invention is to provide an anti-lash gear assembly which reduces gear train noise emissions.

It is another object of the present invention to control load sharing between multiple gear wheels of a scissor gear assembly.

Yet another object is to provide a reliable technique for adjusting gear assemblies to reduce backlash.

Further objects, features, advantages, benefits and aspects of the present invention will become apparent from the drawings and description contained herein.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a front elevational view of an internal combustion engine system of one embodiment of the present invention.

FIGS. 2 and 3 are top plan views of components of an anti-lash gear assembly for the embodiment of FIG. 1.

FIG. 4 is a top plan view of the components of FIGS. 2 and 3 incorporated into the anti-lash gear assembly in an unaligned configuration.

FIG. 5 is a perspective view of the anti-lash gear assembly of FIG. 4 in an aligned configuration.

FIG. 6 is a cross-sectional view of an idler gear and adjustable positioning mechanism along section lines 6—6 of FIG. 1.

FIGS. 7A and 7B are schematic, front elevational views of the system of FIG. 1 at various stages of assembly.

FIGS. 8A—8C are schematic, front elevational views depicting selected operational states of a portion of the system of FIG. 1.

FIG. 9 is a graph illustrating various relationships concerning the operational states shown in FIGS. 8A—8C.

FIG. 10 is an exploded perspective view of an anti-lash gear assembly of an alternative embodiment of the present invention.

FIG. 11A is a top plan view of the anti-lash gear assembly of FIG. 10 in an unaligned configuration.

FIG. 11B is a side elevational view of the anti-lash gear assembly of FIG. 11A.

FIG. 12A is a top plan view of the anti-lash gear assembly of FIG. 10 in an aligned configuration.

FIG. 12B is a side elevational view of the anti-lash gear assembly of FIG. 12A.

FIG. 13 is a section view of a second embodiment of an idler gear adjustment mechanism of the present invention.

FIG. 14 is a cross-sectional view of the gear adjustment mechanism of FIG. 13 taken through section line 14—14.

FIG. 15 is a section view of a third embodiment of a gear adjustment mechanism of the present invention.

FIG. 16 is a cross-sectional view of the gear adjustment mechanism of FIG. 15 taken through section line 16—16.

FIG. 17 is a section view of a fourth embodiment of a gear adjustment mechanism of the present invention.

FIG. 18 is a cross-sectional view of the gear adjustment mechanism of FIG. 17 taken through section line 18—18.

FIG. 19 is a section view of a fifth embodiment of the gear adjustment mechanism of the present invention.

FIG. 20 is a cross-sectional view of the gear adjustment mechanism of FIG. 19 taken through section line 20—20.

FIG. 21 is a section view of a sixth embodiment of the gear adjustment mechanism of the present invention.

FIG. 22 is a cross-sectional view of the gear adjustment mechanism of FIG. 21 taken through section line 22—22.

FIG. 23 is a section view of a seventh embodiment of the gear adjustment mechanism of the present invention.

FIG. 24 is a cross-sectional view of the gear adjustment mechanism of FIG. 23 taken through section line 24—24.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

For the purposes of promoting an understanding of the principles of the invention, reference will now be made to the embodiments illustrated in the drawings and specific language will be used to describe the same. It will nevertheless be understood that no limitation of the scope of the invention is thereby intended. Any alterations and further modifications in the described embodiments, and any further

applications of the principles of the invention as described herein are contemplated as would normally occur to one skilled in the art to which the invention relates.

FIG. 1 depicts internal combustion engine system 20 of the present invention. System 20 includes engine block 22 with a crankshaft 24 shown in phantom. Engine system 20 also includes head assembly 30 connected to block 22. Head assembly 30 includes fuel injector camshaft 32 shown in phantom and valve camshaft 34 shown in phantom. In one embodiment, block 22 and head assembly 30 are configured as a heavy duty, in-line six cylinder diesel engine. The present invention is also applicable to other types of engines as would occur to one skilled in the art.

System 20 includes timing gear train 40. Gear train 40 includes drive gear 42 connected to crankshaft 24. Crankshaft 24 and drive gear 42 have rotational center 44 at the intersection of the crosshairs designated by reference numeral 44. For the figures referenced herein, centers of rotation are depicted by a broken line segment indicative of the corresponding rotational axis when the rotational axis is not perpendicular to the view plane and by crosshairs when the rotational axis is perpendicular to the view plane. Gear 42 rotates with crankshaft 24 during operation of engine system 20 about center 44 to drive the remaining gears of gear train 40.

Gear 42 has teeth 46 which form mesh 48 with lower idler anti-lash gear 50. Gear 50 rotates about shaft 53 having rotational center 54. Shaft 53 is mounted to block 22 by fasteners 55. Bearing 56 provides a rotational bearing relationship between anti-lash gear assembly 58 of gear 50 and shaft 53.

FIGS. 2-5 provide additional details concerning the structure and operation of anti-lash gear assembly 58 of gear 50. Referring to FIG. 2, various details of gear wheel 60 prior to incorporation into gear assembly 58 are shown. Gear wheel 60 includes a hub 63. Web 64 defines seven circumferentially spaced apart apertures 65. Furthermore, for each aperture 65, web 64 defines a fingered edge 65a at one end opposing edge 65b at another end. Aperture 65 and edges 65a, 65b are generally evenly spaced along the circumference of an imaginary circle about center 54. Gear wheel 60 includes a number of circumferentially spaced-apart gear teeth 66 defined by rim 67. Rim 67 is integrally connected to hub 63 by web 64. Adjacent members of gear teeth 66 are generally evenly spaced-apart from one another by gaps 68. Only a few of teeth 66 and gaps 68 are designated to preserve clarity. Each member of gear teeth 66 is generally sized and shaped the same as the others. Similarly, each gap 68 generally has the same size and shape.

Referring to FIG. 3, gear wheel 70 of anti-lash assembly 58 is illustrated. Gear wheel 70 includes hub 73 which is configured to form a rotary bearing relationship with shaft 53 via bearing 56 (see FIG. 1). Hub 63 of gear wheel 60 engages hub 73. The interface between hubs 63 and 73 is adapted to permit rotation of gear wheels 60 and 70 relative to each other. Gear wheel 70 also includes web 74. Tabs 74a project from web 74 in a direction generally perpendicular to the view plane of FIG. 3 and have one side connected to rim 77 to define corresponding recesses 75. At least one tab 74a defines threaded bore 79 therethrough. Bore 79 has a longitudinal axis generally parallel to the view plane of FIG. 3. Web 74 also defines lightening holes 75a each corresponding to one of recesses 75. Tabs 74a and recesses 75 are generally evenly spaced along the circumference of an imaginary circle about center 54.

Wheel 70 includes a number of gear teeth 76 defined by rim 77. Rim 77 is integrally connected to hub 73 by web 74.

Adjacent members of gear teeth 76 are generally evenly spaced-apart from one another by gaps 78. Only a few of teeth 76 and gaps 78 are designated to preserve clarity. Each member of gear teeth 76 is generally sized and shaped the same as the others. Similarly each gap 78 generally has the same shape and size. Preferably, the number of teeth 76 of wheel 70 is the same as the number of teeth 66 of wheel 60.

FIG. 4 defines anti-lash gear assembly 58 in an unaligned configuration commonly encountered prior to preparation for installation in gear train 40. In this configuration, wheels 60 and 70 loosely engage each other so that each aperture 65 of wheel 60 generally overlaps a corresponding recess 75 of wheel 70 to define a number of pockets 80. A number of coil springs 81 are provided each having end 82 opposite end 84. Each spring 81 is positioned in a corresponding one of pockets 80 with end 82 engaging a corresponding tab 74a and end 84 aligning with a corresponding edge 65a. However, ends 84 do not typically engage edges 65a in this configuration.

Assembly 58 also includes adjustment bolt 90 having threaded stem 92 opposing head 94. Stem 92 is shown fully threaded into bore 79 in FIG. 4 with head 94 in contact with corresponding tab 74a. By convention, teeth 66 and 76 are in an "unaligned" position such that teeth 66 overlap gaps 78 defined between teeth 76, and teeth 76 overlap gaps 68 defined between teeth 66. Hub 73 of wheel 70 forms a rotary bearing relationship with hub 63 of wheel 60 so that wheels 60 and 70 are permitted to rotate relative to one another. Head 94 defines contact surface 95 configured to bear against adjacent edge 65b of wheel 60 when wheel 60 is rotated counter-clockwise relative to wheel 70. When wheel 60 is rotated in the clockwise direction relative to wheel 70, spring ends 84 eventually engage corresponding edges 65a. Preferably, each edge 65a defines a finger sized to fit inside the coil of each spring 81 to facilitate proper alignment with wheel 60. When rotated in the clockwise direction with sufficient force, springs 81 are compressed between corresponding edges 65a and tabs 74a, as illustrated in FIG. 5.

FIG. 5 depicts an "aligned" position of gear wheel 60 and 70 reflecting a configuration suitable for installation in gear train 40. When aligned, teeth 76 and 66 are approximately centered over one another as depicted in FIG. 5. Springs 81 are also in a highly compressed condition between edges 65a and tabs 74a to provide a correspondingly high spring force. Adjusting assembly 58 from the configuration of FIG. 4 to the configuration of FIG. 5 is provided by unthreading bolt 90 so that head 94 moves away from bore 79 along stem axis S. As this unthreading continues, surface 95 bears against adjacent edge 65b and springs 81 are compressed between adjacent aligned tabs 74a and edges 65a.

Unthreading of bolt 90 spreads apart the associated tab 74a and edge 65b to rotate wheels 60 and 70 rotate relative to one another and move teeth 66 and 76 past each other. A given tooth of wheel 66 may move into and out of registration with several teeth 76 before reaching the highly biased configuration of FIG. 5 from the unbiased configuration of FIG. 4.

FIG. 5 also depicts face 66a of each tooth 66 of wheel 60 a few of which are depicted. Each tooth 76 of wheel 70 similarly has a face 76a, a few of which are depicted. Width W60 represents the width of a typical face 66a. Similarly, width W70 represents the width of a typical face 76a. Preferably, width W60 is less than width W70. More preferably, width W70 is at least about 50% greater than width W60. Most preferably, width W70 is at least about twice width W60.

Referring collectively to FIGS. 4 and 5, anti-lash gear wheel assembly 58 is constructed by providing wheel 70 and mounting one of springs 81 to align with bore 79. Bolt 90 is threaded into bore 79 so that head 94 contacts the associated tab 74a. The remaining springs 81 are placed in recesses 75 of wheel 70. Wheel 60 is placed over wheel 70 to define corresponding pockets 80 generally evenly spaced along imaginary circle 86 (shown in phantom in FIG. 4). Edges 65a align with ends 84 of corresponding springs 81.

Prior to mounting assembly 58 on shaft 53, it is preferred that teeth 66 and 76 be aligned. To provide this alignment, bolt 90 is partially unthreaded from bore 79 so that head 94 contacts adjacent edge 65b of wheel 60 and correspondingly compresses springs 81. In response, teeth 66, 76 move past one another. Unthreading of bolt 90 continues this motion until the aligned position of FIG. 5 is generally reached. As a result, wheel 60 is separated from wheel 70 along stem axis S by distance D as illustrated in FIG. 5. Notably, a portion of stem 92 of bolt 90 remains threaded in bore 79 in both the unaligned position of FIG. 4 and in the aligned position of FIG. 5. In other embodiments, more than one or all of tabs 74a may be adapted to define a bore 79 suitable for engagement by bolt 90. Similarly, multiple bolts 90 may be employed with embodiments having multiple bores 79.

Once teeth 66 and 76 are aligned in the configuration of FIG. 5, assembly 58 is mounted to shaft 53 via bearing 56. When so mounted, the aligned teeth 66, 76 form mesh 48 with teeth 46 of drive gear 42. However, mesh 48 typically has a significant amount of lash when teeth 66, 76 are forcibly aligned by the extension of bolt 90. To take-up this lash with gear 50, wheels 60 and 70 are preferably permitted to rotate relative to one another under the influence of the bias provided by compressed springs 81. Threading bolt 90 back into bore 79 once assembly 58 is mounted to form mesh 48 with drive gear 42 permits this rotation. As a result, the spring bias offsets teeth 66 and 76 from one another to generally occupy the entire space between adjacent teeth 46 participating in mesh 48. Notably, mesh 48 does not permit teeth 66, 76 to return to the unloaded position of the FIG. 4 configuration.

Each pair of initially aligned teeth 66, 76 operate collectively as a composite tooth with a variable effective size or "thickness" dependent upon the space between mating teeth 46. By varying in thickness, these composite teeth may reduce, or even effectively eliminate, backlash in mesh 48. To conclude installation of assembly 58, bolt 90 should be tightened down so that head 94 bears against the associated tab 74a. Bolt 90 is preferably carried by wheel 70 throughout the adjustment process and utilization of assembly 58 as part of gear 50.

Preferably, wheel 60 and 70 are machined from a metallic material suitable for long-term use in a diesel engine timing gear train. It is also preferred that bolt 90 and springs 81 be selected from compatible materials suitable for long term use in a diesel engine environment. Nonetheless, in other embodiments, different materials may be used as would occur to one skilled in the art.

Although gear 50 is illustrated in FIG. 1 as an idler gear, in other configurations it may be configured as a driving gear, a driven gear, or otherwise adapted or modified as would occur to one skilled in the art. In all these forms, gear 50 may be considered to be a novel type of "scissor gear."

Referring back to FIG. 1, gear 50 participates in gear train 40 to form mesh 96 with idler gear 100. Idler gear 100 rotates about rotational center 104 and defines circumferential teeth 106 spaced apart by gaps 108 to form mesh 96 with gear 50.

Referring additionally to FIG. 6, further details concerning idler gear 100 are provided. Idler gear 100 includes rim 107 defining teeth 106 integrally connected to web 114. Web 114 defines lightening holes 116. Web 114 is also integrally connected to hub 118 which, as shown in the cross-sectional view of FIG. 6, has slightly less thickness along the rotational axis corresponding to center 104 than rim 107. Cylindrical bushing 119 provides a rotational bearing surface between shaft 103 and hub 118. Shaft 103 defines four passages 105 used to mount idler gear 100 to block 22.

Mounting of idler gear 100 is provided by adjustable positioning mechanism 120. Mechanism 120 includes a mounting plate 130 which is positioned between shaft 103 of idler gear 100 and block 22. Notably, plate 130 is configured to provide clearance with hub 118 of idler gear 100 so that idler gear 100 may freely rotate about shaft 103.

Idler gear 100 and mounting plate 130 are positioned between block 22 and retaining plate 140. Retaining plate 140 includes mounting holes 145 which are generally aligned with mounting passages 105 of shaft 103, mounting passages 135 of plate 130, and threaded bores 25 of block 22. Notably passages 105 have a larger dimension along an axis perpendicular to the rotational axis of gear 100 than passages 135, holes 145, and bores 25. Idler gear 100 is secured between plates 130 and 140 by inserting cap screw fasteners 150 through holes 145, passages 105, and passages 135 and threading the end of threaded stems 152 into bores 25. Fasteners 150 each have head 154 opposing threaded stem 152. Head 154 is sized to contact retaining plate 140 when stems 152 are fully threaded into bores 25 to clamp plate 140 against shaft 153 and to clamp shaft 153 against plate 130.

In operation, mechanism 120 is configured to position idler gear 100 relative to a planar region that is preferably parallel to the view plane of FIG. 1 and perpendicular to the view plane of FIG. 6. Within this region, gear 100 may be positioned with two degrees of freedom as symbolized by the X and Y directional arrows of FIG. 1.

To mount idler gear 100, mounting plate 130 is first secured to block 22 using fasteners (not shown) in a conventional manner so that passages 135 align with bores 25. Once plate 130 is secured to block 22, idler gear 100 is located on plate 130 so that passages 105 overlap passages 135. Next, retaining plate 140 is placed over shaft 103 to locate holes 145 over corresponding passages 105 and 135, and bores 25. Fasteners 150 are then each placed through an aligned hole 145, passage 105, and passage 135 and loosely threaded into a corresponding bore 25. Preferably, fasteners 150 are initially threaded into bores 25 an amount sufficient to contact plate 140 and yieldingly hold idler gear 100 in position. In this configuration, the position of idler gear 100 relative to the planar region symbolized by the X and Y directional arrows may be selected within the range permitted by the clearance of fasteners 150 in passages 105. Once an X-Y position is selected, fasteners 150 are tightened down to secure idler gear 100 and mechanism 120.

Teeth 106 of idler gear 100 form mesh 196 with anti-lash gear 200. Gear 200 is mounted to fuel injector camshaft 32 of head assembly 30 and is configured to rotate about rotational center 204. Gear 200 is preferably configured similar to gear 50 having composite gear tooth pairs represented by reference numeral 266. Furthermore, springs 281 of gear 200 are shown configured in a manner similar to springs 81 of gear 50, although fewer in number (three being shown). Likewise an installation adjustment bolt 290 is shown. This adjustment bolt may function for installation

purposes similar to bolt **90** of gear **50**. Gear **50**, gear **200**, or both may utilize belleville washers to provide a spring bias either with or without coil springs.

Gear **200** forms mesh **296** with mating gear **300**. Mating gear **300** is attached to valve camshaft **34** to rotate about rotational center **304**. Gear **300** defines teeth **306** which interface with tooth pairs **266** of gear **200** to form mesh **296**.

In operation, drive gear **42** rotates with crankshaft **24** to turn gear **50**. In response, gear **50** turns idler gear **100** via mesh **96**. Idler gear **100** drives gear **200** via mesh **196** to regulate timing of fuel injectors (not shown) for engine system **20** by rotating fuel injector camshaft **32**. Furthermore, gear **200** drives mating gear **300** via mesh **296** to rotate valve camshaft **34** therewith to time engine valves (not shown) for head assembly **30**. Thus, gear train **40** turns camshafts **32** and **34** of head assembly **30** in response to rotation of crankshaft **24** to control timing of engine system **20**.

In other embodiments, different quantities and arrangements of gears in gear train **40** may be utilized as would occur to one skilled in the art. In one alternative embodiment, a conventional scissor gear may be used in place of gear **50**, gear **200**, or both. In still other embodiments an idler gear with an adjustable positioning mechanism may not be required.

In one embodiment of gear train **40**, the number of teeth **46** is about **48** for drive gear **42**; the number of teeth **66**, **76** is about **70** for gear wheels **60**, **70**, respectively; the number of teeth **106** for adjustable idler gear **100** is about **64**; the number of composite teeth **266** for gear **200** is about **76** and the number of teeth **306** is about **76** for gear **300**. Furthermore, for this configuration, gears **42**, **50**, **100**, **200**, **300** are of a spur gear configuration, are made from metallic materials suitable for long term use with internal combustion engines, and have generally parallel rotational axes which perpendicularly intersect the view plane of FIG. 1.

Having described selected structural and operational features of system **20**, certain aspects concerning the assembly of system **20** are next described in connection with the schematic representations of FIGS. 7A and 7B. In FIGS. 7A and 7B, reference numerals schematically depict structure identified by like reference numerals in FIGS. 1–6; however, gear meshes have been enlarged to emphasize selected features of the present invention. FIG. 7A illustrates an intermediate assembly stage of drive train **40**. In this stage, drive gear **42** has been previously mounted to rotate about center **44** in the direction indicated by arrow **R1**. Similarly, mating gear **300** has been mounted to rotate about center **304** in the direction indicated by arrow **R5**.

After gears **42** and **300** have been mounted, gears **50** and **200** are mounted to form mesh **48** between gears **42** and **50**, and mesh **296** between gears **200** and **300**. The formation of meshes **48**, **296** determines the effective composite tooth size of corresponding pairs of teeth for gears **50** and **200** as they occupy gaps between teeth **46** and **306** of gears **42** and **300**, respectively. For gear **50**, teeth **76** of wheel **70** are represented by dashed lines, and teeth **66** of wheel **60** are represented by solid lines for illustrative purposes. The effective circular thickness **T50** of one composite tooth pair of gear **50** is also shown. This composite circular thickness is determined along a pitch circle of gear **50** for mesh **48**. Notably, in the absence of idler gear **100**, thickness **T50** is defined by the mating gap of teeth **46** of gear **42**.

For mesh **296**, gear **200** forms composite teeth pairs **266**. Each pair **266** has a member represented by a dashed line and a member represented by a solid line to enhance clarity.

The effective circular tooth thickness of one composite tooth pair **266** is shown as circular thickness **T200** relative to a pitch circle for gear **200**.

Arrows **R4**, **R5** indicate the rotational direction in which gears **200**, **300** are driven, respectively. Also indicated are mounting bores **25** of engine block **22** for reference.

Having defined the composite circular thicknesses **T50** and **T200**, idler gear **100** is installed to form mesh **96** with gear **50** and mesh **196** with gear **200** as depicted in FIG. 7B. The tooth thicknesses **T50** and **T200** are typically different corresponding to a difference in the amount of backlash in meshes **48** and **296**. By using mechanism **120** to adjust the X-Y position of rotational center **104** relative to fixed rotational centers **54** and **204**, idler gear **100** may be located to optimally mesh with the pre-defined tooth sizes of gears **50** and **200** despite any lash difference. Fasteners **150** of mechanism **120** are illustrated in FIG. 7B for reference.

The positional adjustment of idler gear **100** relative to the other gears results in significant control over the amount of backlash in meshes **96** and **196**. When the backlash difference resulting from different **T50** and **T200** widths is within a certain range, backlash may be reduced, or even effectively eliminated, through proper placement of idler gear **100** along a planar region perpendicular to the rotational axes of the meshing gears.

Notably, while the preferred embodiment presents two meshes **96**, **196** with idler gear **100**, in other embodiments this assembly method may be practiced to control backlash for a different quantity of meshing gears. For example, this assembly technique finds application in gear trains having only three gears oriented similar to gears **42**, **50**, and **100**.

Referring to FIGS. 8A–8C, selected operational states of gears **42**, **50**, and **100** are schematically depicted with reference numerals representing structure designated by like numerals in FIGS. 1–6; however, fewer and larger teeth are schematically illustrated in these figures to emphasize various features. Referring to FIG. 8A, gears **42**, **50**, **100** are in a static (motionless) state relative to each other. Referring to mesh **48**, imaginary pitch circles **C1**, **C2**, **C3** are represented by dashed lines for gears **42**, **50**, **100**, respectively. The circular thickness **T50a** of a pair of gear teeth **76**, **66** of gear **50** is shown as an arc along the companion pitch circle **C2**. Arrows **DF1** represent the forces counter-acting the bias of gear **50** for the depicted static condition in FIG. 8A. The static reaction forces of gear **100** are shown by arrows **RF1**. Also depicted is the circular thickness **T60** of a selected tooth **66**, and the circular thickness **T70** of a selected tooth **76**. It is preferred that circular thickness **T60** be nominally less than circular thickness **T70** for each tooth **60**, **70**, respectively. In one preferred embodiment, **T60** is at least about two thousandths (0.002) of an inch less than **T70**. More preferably, this difference is at least about four thousandths (0.004) of an inch. Most preferably, this difference is in a range of about two to six thousandths (0.002–0.006) of an inch.

In FIG. 8B, drive gear **42** is rotating in the direction indicated by arrow **R1** to provide a resultant drive force represented by arrow **DF2**. In response, gear **50** is rotating in the direction indicated by arrow **R2** and gear **100** is rotating in the direction indicated by arrow **R3**. The resultant reaction force presented by gear **100** is represented by arrow **RF2**. The resultant forces **DF2** and **RF2** are of sufficient intensity to partially overcome the spring bias, causing compression of springs **81** of gear **50**. As a result, the circular thickness **T50b** of the composite pairs of teeth of gear **50** decreases relative to thickness **T50a** (**T50b** is less

than T50a). As the magnitude of the force transmitted from drive gear 42 increases, gear teeth 66, 76 continue to approach alignment.

In FIG. 8C, the resultant driving force DF3 of gear 42 and reaction force RF3 of gear 100 compresses springs 81 by an amount sufficient to align gear teeth 66 and 76. When so aligned, composite thickness T50c results. T50c is less than both T50a and T50b, and is generally equal to the circular thickness T70 of teeth 76. Springs 81 are generally fully compressed in the FIG. 8C configuration; storing energy generally equivalent in amount to springs 81 in the configuration of FIG. 5.

The smaller circular thickness of teeth 66 compared to teeth 76 (T60<T70) prevents loading of teeth 66 beyond the load provided by the compressed springs of FIG. 8C. In contrast, teeth 76 bear any load in excess of the spring load. Limiting the load on teeth 66 to the spring bias generally reduces reverse bending loads commonly resulting from random dimensional differences of tooth pairs having each member nominally sized to the same circular thickness. Preferably, the wider tooth face W70 of each tooth 76 is selected to bear the higher driving loads in excess of the spring bias; however, the total width increase (W60+W70) for gear is typically less than the width increase required to withstand reverse bending loads by a scissor gear that has the same nominal circular thickness for all teeth.

FIG. 9 graphically represents the typical effect of reduced circular thickness T60 compared to circular thickness T70 with load lines 400. The dashed line 400 represents gear wheel 60 and the solid line 400 represents gear wheel 70. Horizontal segments 410 correspond to the pre-loaded bias of gear 50 under the static conditions of FIG. 8A. Sloped segments 420 correspond to the loading of teeth 66, 76 between the static condition of FIG. 8A and the aligned position of FIG. 8C. FIG. 8B represents one point along segments 420. Once loading compresses springs 81 to align teeth as illustrated in FIG. 8C, the loading on teeth 66 of gear wheel 60 flattens to the maximum load of springs 81 as indicated by segment 430. At the same time, the thicker face W70 of teeth 76 bears the high intensity loading as indicated by sloped segment 440. By allowing wheel 70 to handle the high loads and limiting loading of wheel 60 with the circular thickness differential (T70-T60), reverse bending loads are typically reduced.

It has been found that much of the unpleasant noise, such as the "hammering" sounds associated with heavy-duty diesel engines, is due to high impact noise from gear trains associated with those engines. An unexpectedly dramatic change in sound quality is experienced, typically including a reduction in overall noise intensity, when a relatively high bias torque is provided by a scissor gear participating in the gear train. As used herein, "bias torque" is the magnitude of the torque provided by a spring-biased scissor gear assembly. The bias torque is determined as the magnitude of the cross product of the vectors corresponding to a radial distance from the rotational center of the gear to the teeth and the force acting tangential to a circle corresponding to the radius. Typically, the bias torque varies as a function of the amount of loading of the scissor gear bias. Preferably, the bias torque is at a maximum when the gear teeth are generally aligned in opposition to the bias. For the aligned configuration of teeth 66, 76 in FIG. 5, a radial vector T and a force vector F are illustrated which may be used to determine bias torque for assembly 58.

It has been found that a maximum bias torque of at least 100 foot-pounds (ft-lbs) provides improved gear train noise

character and intensity. More preferably, a maximum bias torque of at least about 200 ft-lbs is provided. Most preferably, a maximum bias torque of at least about 500 ft-lbs is provided. In one most preferred embodiment, gear 50 is configured with a maximum bias torque of about 700 ft-lbs, and gear 200 is configured with a maximum bias torque of about 200 ft-lbs. In many instances, the bias torque of the present invention obviates the need to use expensive enclosures and panels to mute unpleasant noise.

FIG. 10 provides an exploded perspective view of anti-lash gear assembly 558 about rotational center 554 of an alternative embodiment of the present invention. Assembly 558 includes gear wheel 560 with splines 561 defined by inner cylindrical surface 564 of hub 563. Hub 563 defines opening 563a therethrough. Splines 561 are of the helical type oriented about center 554 and inclined relative to the rotational axis of wheel 560. Hub 563 is integrally connected to web 564. A number of circumferentially disposed teeth 566 are defined by rim 567 which is also integrally connected to web 564. Teeth 566 are generally evenly spaced apart from each other about center 554 and each have generally the same size and shape. Between adjacent teeth 566 are gaps 568 which are also generally evenly spaced apart from one another and have generally the same shape and size. Web 564 of wheel 560 defines two opposing apertures 569 therethrough.

Assembly 558 also includes wheel 570. Wheel 570 includes splines 571 defined by cylindrical outer surface 572 of hub 573. Splines 561 are of the helical type oriented about center 554 and inclined relative to the rotational axis of wheel 570. Splines 571 are inclined in generally the same manner as splines 561 to mate therewith. Hub 573 is configured to fit within opening 563a of hub 563 to mate splines 561 and 571. Hub 573 defines opening 573a surrounded by inner cylindrical surface 574 for establishing a rotational bearing relationship with a mounting shaft. Wheel 570 also includes web 574 integrally connected to hub 573. Teeth 576 are defined by rim 577 which is integrally connected to web 574. Teeth 576 are generally evenly spaced apart from one another about rotational center 554 and each have generally the same size and shape. Teeth 576 define gaps 578 therebetween. Gaps 578 are generally evenly spaced apart from one another and each have generally the same size and shape. Collectively, hub 573, web 574, and rim 571 define a cylindrical recess 575. Web 564 defines two opposing threaded recesses 579 each corresponding to one of apertures 569.

Coil springs 580 are each placed in recess 575 and are generally evenly spaced apart from one another about center 554 between hub 573 and rim 577. Adjustment devices 590a, 590b are included which each have adjustment bolt 590 with threaded stem 592. Stem 592 has end 593 opposing head 594. Devices 590a, 590b each include washer 596 configured for passage of stem 592 therethrough. In contrast, head 594 is sized so that it will not pass through washer 596. Also, the outer diameter of washer 596 is dimensioned so that it will not pass through aperture 569. Aperture 569 is sized to provide ample clearance for stem 592, permitting selective positioning of stem 592 therein. Threaded recesses 579 are each configured for engagement by a corresponding one of stems 592.

Referring to FIG. 11A, an unaligned position of assembly 558 is illustrated that shows teeth 566 and 576 of wheels 560 and 570, respectively, out of register similar to the embodiment illustrated in FIG. 4. Referring additionally to FIG. 11B, a side elevational view of assembly 558 in the unaligned configuration is illustrated. Splines 561 of wheel

560 engage splines **571** of wheel **570**. For each device **590a**, **590b**, stems **592** have corresponding longitudinal stem axes **S1**, **S2**. Stems **592** are inserted through corresponding washers **596** and apertures **596** to initially engage a corresponding threaded recess **579**. Springs **580** are not substantially compressed in the configuration of FIGS. **11A** and **11B**.

Referring additionally to FIGS. **12A** and **12B**, a perspective view and a side elevational view of assembly **558** in an aligned configuration are illustrated, respectively. This aligned configuration generally corresponds to the aligned configuration of assembly **58** illustrated in FIG. **5**. To provide alignment of assembly **558**, stems **592** of adjustment devices **590a**, **590b** are further threaded into recesses **579** to compress springs **580** between wheel **560** and **570**. As springs **580** are compressed, the inclines of mating splines **561**, **571** provide a ramping action that generally converts the translational motion of devices **590a**, **590b** to a rotational motion of wheels **560**, **570**. As stems **592** of devices **590a**, **590b** are unthreaded, the compressed springs **580** provide a force which rotates wheels **560** and **570** in the opposite direction due to the engagement of splines **561**, **571**. Assembly **558** is configured so that teeth **566** and **576** are generally aligned when stems **592** are fully threaded into recesses **579**. This aligned orientation of assembly **558** is also preferably configured to provide a selected maximum bias torque. The distance wheels **560** and **570** occupy along stem axes **S1** and **S2** changes from **D1** for the unaligned position shown in FIG. **11B** to **D2** for the aligned position illustrated in FIG. **12B**, where **D1** is greater than **D2**. Notably, **D2** is the minimum distance occupied by wheels **560**, **570** of assembly **558** along stem axes **S1**, **S2**. Thus, wheels **560**, **570** rotate relative to each other in accordance with the distance occupied by the wheels **560**, **570** along the rotational axis corresponding to center **554**.

Preferably the number of teeth **566** is the same as the number of teeth **576**. It is also preferred that the number of helical splines **561**, **571** be the same as the number of teeth **566**, **576**, respectively. Identical quantities of teeth and splines simplifies assembly by avoiding the need to index splines **571**, **561** to assure that alignment of teeth **566** and **576** coincides with high spring compression. In other embodiments, aperture **569** may be configured as a non-circular opening as opposed to the generally circular opening illustrated in FIG. **10**. In one alternative embodiment, aperture **569** is configured as an arcuate slot with a bend radius extending from center **554**.

Splines **561**, **571** may be provided in different locations besides hubs **563**, **573**. By way of non-limiting example, arcuate slots defined by one wheel may have an inner surface defining splines configured to mate with splines defined by a flange extending from the other wheel into these slots. Notably, one or more segments of mating spines oriented about the rotational axis are capable of providing the relative rotation of the gear wheels without needing to encircle the axis.

Similar to the embodiment of assembly **58**, assembly **558** provides an alignment device which provides for selectively aligning teeth of two gear wheels of an anti-lash gear assembly by opposing the spring bias of the assembly. Stems **592** are tightened down to provide the aligned configuration of FIGS. **12A** and **12B** for installation. Once assembly **558** is meshed with another gear, gear **42** for example, stem **592** of each device **590a**, **590b** is loosened to permit relative rotation of wheels **560** and **570** to take-up lash of the mating gear. This loosened position would appear similar to the configuration of FIGS. **11A** and **11B**, but would preferably provide clearance between head **594** and washer **596** of each

bolt **590** to accommodate changing lash conditions of a corresponding mesh. In one embodiment, devices **590a**, **590b** are removed once assembly **558** is installed in a mesh with another gear. This embodiment relies on the mesh to oppose the bias.

Each assembly **58**, **558** is configured with an adjustment device having a threaded stem coupled to one wheel that extends along a stem axis. These devices further include a head coupled to the stem and configured for adjustable positioning relative to the wheel. Generally, assemblies **58** and **558** may be configured to be interchangeable with regard to other features of the present invention. Furthermore, assembly **58** or **558** may be adapted for use with anti-lash gear **200**. In other embodiments of assemblies **58**, **558**; bolts **90**, **590** may be replaced by a threaded stem fixed to one of the wheels with a nut threaded thereon to provide a movable head. This nut is positioned along the stem to selectively engage the other wheel. In still other embodiments of the present invention, neither anti-lash assembly may be utilized. Indeed in some alternative embodiments of the present invention, a conventional scissor gear assembly may be employed.

FIGS. **13** and **14** depict adjustable idler gear mechanism **600a** of the present invention. Mechanism **600a** is illustrated as a substitute for idler gear **100** and mechanism **120** of system **20** with like reference numerals representing like features. Mechanism **600a** includes gear **603a**, mounting shaft **655a**, and mounting plate **630a** coupled to block **22** as previously described in connection with system **20**. Gear **603a** includes gear wheel **613a** and mounting shaft **655a**. Gear wheel **613a** has rim **616a** defining teeth **615a**. Rim **616a** is integrally connected to web **617a**. Web **617a** is integrally connected to hub **618a** which engages cylindrical bushing **619a** to provide a rotational bearing relationship in the manner as previously described for idler gear **100**.

Mounting shaft **655a** has positioning pin **601a** protruding from mounting surface **667a**. Pin **601a** is shown engaging slot **602a** defined by mounting surface **637a** of mounting plate **630a**. Mounting surfaces **667a** and **637a** are also shown in engagement. Pin **601a** is fixed relative to shaft **655a** and is generally centered relative to the rotational axis that coincides with rotational center **650a** of gear **603a**. Pin **601a** may be an integral part of shaft **655a** or connected thereto using any suitable means as would occur to those skilled in the art. Slot **602a** follows a limited pathway represented by a line segment with the reference numeral **660a** in FIG. **14**. Collectively, pin **601a** and slot **602a** provide a guide **625a** to position center **650a** along pathway **660a**. **660a** may follow a generally straight path, as represented by the solid line segment, or alternatively, it may follow a curvilinear path **660a'** as shown in exaggerated form by the dashed line segment of FIG. **14**. Plate **630a** is fixed relative to block **22** using any suitable means that would occur to those skilled in the art. The location of plate **630a** is selected to provide a predetermined relationship of pathway **660a**, **660a'** relative to rotational center **54** of gear **50**.

In one preferred assembly process, rotational centers **54** and **204** of gears **50** and **200**, respectively, are predetermined. Mounting plate **630a** is positioned to establish that the distance separating rotational center **54** from any given point along pathway **660a** stays within a predetermined range. This range corresponds to an acceptable range of backlash variation between gears **50** and **603a** as more fully described hereinafter. After gears **50** and **200** are mounted, gear **603a** is positioned to adjustably intermesh therewith and pin **601a** is received in slot **602a**. Pin **601a** slidably

engages slot **602a** to provide guide **625a**, and thereby facilitate adjustment of rotational center **650a** of gear **603a** along pathway **660a**. For any position of pin **601a** along slot **602a**, the spacing between rotational centers **54** and **650a** is maintained to provide an acceptable minimal degree of backlash for the mesh between gears **603a** and **50** in accordance with the predetermined spatial relationship between pathway **660a** and rotational center **54**. For the generally straight pathway configuration designated by reference numeral **660a**, it is preferred to orient pathway **660a** to be approximately parallel to tangent line **610a**; where line **610a** is tangent to pitch circle **612a** for gear **50** and pitch circle **614a** for gear **603a**. This orientation generally minimizes the amount of backlash variation between gear **50** and gear **603a** as pin **601a** is moved along pathway **660a**.

Further, this technique of determining the position of assembly **600a** relative to gears **50**, **200** is less prone to error, being more “foolproof” than existing methods to minimize backlash of two or more meshes of the same gear. Also, the structure of guide **625a** is less complicated and time consuming to adjust compared to conventional approaches.

Typically, it is easier to manufacture guide **625a** to define a generally straight pathway **660a** than a predetermined curved pathway **660a'**; however, the degree of accuracy available when matching the backlash of the meshes of both gears **50** and **200** may be generally less for pathway **660a** than curved path **660a'**. Thus, in an alternative preferred embodiment, slot **602a** follows curved pathway **660a'**. The curvature of pathway **660a'** is preferably selected to provide generally constant spacing from rotational center **54** as pin **601a** is moved therealong. For this preferred embodiment, pathway **660a'** has a generally constant radius of curvature originating from rotational center **54**, such that pathway **660a'** is a concentric arc relative to pitch circle **612a** of gear **50**.

Pathways **660a** and **660a'** are also oriented to provide a substantially greater degree of spacing variation and corresponding backlash variation with respect to gear **200**. Therefore, pin **601a** may be moved along pathway **660a** or **660a'** to determine a position which minimizes backlash between gear **200** and **603a** while not exceeding a desired level of backlash between gear **50** and **603a**. Once a position is selected, shaft **655a** is fixed relative to mounting plate **630a** and block **22**. This fixation may be accomplished with one or more fasteners, such as with bolts, screws, or rivets; a bonding process, such as soldering, brazing, welding, or adhesively joining; or through such other means as would occur to one skilled in the art. In another embodiment, pin **601a** protrudes from the plate **630a** and the slot **602a** is defined by the shaft **655a** to provide guide **625a** with comparable adjustment capability.

FIGS. **15** and **16** depict another idler gear adjustment mechanism **600b** of the present invention. Mechanism **600b** is illustrated as a substitute for idler gear **100** and mechanism **120** of system **20** with like reference numerals representing like features. Mechanism **600b** includes gear **603b**, mounting shaft **655b**, and mounting plate **630b** coupled to block **22** as previously described in connection with system **20**. Gear **603b** includes gear wheel **613a**, as described in connection with mechanism **600a**. Likewise, gear **603b** includes cylindrical bushing **619b** to provide a rotational bearing relationship between gear wheel **613a** and shaft **655b**.

Adjustment mechanism **600b** has a guide **625b** including positioning pins **601b** and slot **609b**. Mounting shaft **655b** has two positioning pins **601b** protruding from mounting surface **667b**. Pins **601b** are shown engaging slot **609b**

defined by mounting surface **637b** of mounting plate **630b**. Mounting surfaces **667b** and **637b** are also shown in engagement. Pins **601b** are fixed relative to shaft **655b** in an off-center relationship relative to the rotational axis that coincides with rotational center **650b** of gear **603b**. Pins **601b** may be an integral part of shaft **655b** or connected thereto using any suitable means as would occur to those skilled in the art. Slot **609b** follows a limited pathway represented by a line segment with the reference numeral **660b** in FIG. **16**. Plate **630b** is fixed relative to block **22** using any suitable means that would occur to those skilled in the art.

In one preferred assembly process, plate **630b** is mounted to block **22** to establish a predetermined spatial relationship between pathway **660b** and rotational center **54** of gear **50**. Gear **603b** is mounted to adjustably intermesh with gears **50** and **200** by slidably engaging pins **601b** in slot **609b**. For any position of pins **601b** along slot **609b**, the spacing between rotational centers **54** and **650b** is maintained to provide an acceptable minimal degree of backlash for the mesh between gears **603b** and **50** in accordance with the predetermined spatial relationship between pathway **660b** and rotational center **54**. For the illustrated embodiment, pathway **660b** is represented by a generally straight segment that is generally parallel to tangent line **610b**; where line **610b** is tangent to pitch circle **612b** for gear **50** and pitch circle **614b** for gear **603b**. In an alternative embodiment, pathway **660b'** is represented by a dashed line, and is curved and configured in the manner described in connection with pathway **660a'** of mechanism **600a**. The curvature is exaggerated to more clearly depict pathway **660b'**.

Pathway **660b** is also oriented to provide a substantially greater degree of spacing variation and corresponding backlash variation with respect to gear **200**. Therefore, pins **601b** may be moved along pathway **660b** to determine a position which minimizes backlash between gear **200** and **603b** while not exceeding a desired level of backlash between gears **50** and **603b**. Once a position is selected, shaft **655b** is fixed relative to mounting plate **630b** and block **22** in a manner comparable to that described for plate **630a**.

FIGS. **17** and **18** depict another idler gear adjustment mechanism **600c** of the present invention. Mechanism **600c** is illustrated as a substitute for idler gear **100** and mechanism **120** of system **20** with like reference numerals representing like features. Mechanism **600c** includes gear **603c**, mounting shaft **655c**, and mounting plate **630c** that have been coupled to block **22** previously described in connection with system **20**. Gear **603c** includes gear wheel **613a** described in connection with mechanism **600a**. Likewise, gear **603c** includes cylindrical bushing **619c** to provide a rotational bearing relationship between gear wheel **613c** and shaft **655c**.

Adjustment mechanism **600c** has a guide **625c** including pins **601c** and shoulder **604c**. Mounting plate **630c** has two positioning pins **601c** protruding from mounting surface **637c**. Pins **601c** are shown engaging guide shoulder **604c** defined by mounting surface **667c** of shaft **655c**. Mounting surfaces **667c** and **637c** are also shown in engagement. Pins **601c** are fixed relative to plate **630c** in an off-center relationship relative to the rotational axis that coincides with rotational center **650c** of gear **603c**. Pins **601c** may be an integral part of plate **630c** or connected thereto using any suitable means as would occur to those skilled in the art. Shoulder **604c** follows a pathway represented by a line segment with the reference numeral **660c** in FIG. **18**. Plate **630c** is fixed relative to block **22** using any suitable means that would occur to those skilled in the art.

In one preferred assembly process, plate **630c** is mounted to block **22** to establish a predetermined spatial relationship

between pathway 660c and rotational center 54 of gear 50. Gear 603c is mounted to adjustably intermesh with gears 50 and 200 by sliding pins 601c against shoulder 604c. For any position of pins 601c along shoulder 604c, the spacing between rotational centers 54 and 650c is maintained to provide an acceptable minimal degree of backlash for the mesh between gears 603c and 50 in accordance with the predetermined spatial relationship between pathway 660c and rotational center 54. For the illustrated embodiment, pathway 660c is represented by a generally straight segment that is generally parallel to tangent line 610c; where line 610c is tangent to pitch circle 612c for gear 50 and pitch circle 614c for gear 603c. In an alternative embodiment, curved pathway 660c' may be utilized as represented by the dashed line in FIG. 18. Preferably, pathway 660c' is curved in the manner described for pathway 660a' of the mechanism 600a. The curvature of pathway 660c' is exaggerated to more clearly depict it in FIG. 18.

Pathway 660c is also oriented to provide a substantially greater degree of spacing variation and corresponding backlash variation with respect to gear 200. Therefore, pins 601c may be moved along pathway 660c to determine a position which minimizes backlash between gear 200 and gear 603c while not exceeding a desired level of backlash between gear 50 and gear 603c. Once a position is selected, shaft 655c is fixed relative to mounting plate 630c and block 22 in a manner comparable to that described for plate 630a.

FIGS. 19 and 20 depict another idler gear adjustment mechanism 600d of the present invention. Mechanism 600d is illustrated as a substitute for idler gear 100 and mechanism 120 of system 20 with like reference numerals representing like features. Mechanism 600d includes gear 603d, mounting shaft 655d, and mounting plate 630d that have been coupled to block 22 previously described in connection with system 20. Gear 603d includes gear wheel 613a described in connection with mechanism 600a. Likewise, gear 603d includes cylindrical bushing 619d to provide a rotational bearing relationship between gear wheel 613a and shaft 655d.

Adjustment mechanism 600d has a guide 625d including positioning pins 601d and guide walls 604d. Mounting plate 630d has two positioning pins 601d projecting from mounting surface 637d that are oppositely disposed relative to rotational center 650d. Pins 601d each engage a corresponding one of a pair of guide walls 604d defined by surface 667d of shaft 655d. Guide walls 604d are generally parallel to each other and are generally opposite each other. Surfaces 667d and 637d are also shown in engagement. Pins 601d are fixed relative to plate 630d in a generally symmetric relationship about rotational center 650d of gear 603d. Pins 601d may be an integral part of plate 630d or connected thereto using any suitable means as would occur to those skilled in the art. Walls 604d generally correspond to a pathway represented by a line segment with the reference numeral 660d in FIG. 20. Plate 630d is fixed relative to block 22 using any suitable means that would occur to those skilled in the art.

In one preferred assembly process, plate 630d is mounted to block 22 to establish a predetermined spatial relationship between pathway 660d and rotational center 54 of gear 50. Gear 603d is mounted to adjustably mesh with gears 50 and 200 by sliding each pin 601d against a corresponding wall 604d. For any position of pins 601d along walls 604d, the spacing between rotational centers 54 and 650d is maintained to provide an acceptable minimal degree of backlash for the mesh between gears 603d and 50 in accordance with the predetermined spatial relationship between pathway

660d and rotational center 54. For the illustrated embodiment, pathway 660d is represented by a generally straight segment that is generally parallel to tangent line 610d; where line 610d is tangent to pitch circle 612d for gear 50 and pitch circle 614d for gear 603d. In an alternative embodiment, the pathway is represented by a dashed line in FIG. 20 with reference numeral 660d' and is curved in the manner described in connection with pathway 660a' of mechanism 600a. The curvature of pathway 660d' is exaggerated in FIG. 20 to more clearly depict it.

Pathway 660d is also oriented to provide a substantially greater degree of spacing variation and corresponding backlash variation with respect to gear 200. Therefore, pins 601d may be moved along pathway 660d to determine a position which minimizes backlash between gear 200 and gear 603d while not exceeding a desired level of backlash between gear 50 and gear 603d. Once a position is selected, shaft 655d is fixed relative to mounting plate 630d and block 22 in a manner comparable to that described for plate 630a.

FIGS. 21 and 22 depict another idler gear adjustment mechanism 600e of the present invention. Mechanism 600e is illustrated as a substitute for idler gear 100 and mechanism 120 of system 20 with like reference numerals representing like features. Mechanism 600e includes gear 603e, mounting shaft 655e, and mounting rail 630e coupled to block 22 as previously described in connection with system 20. Gear 603e includes gear wheel 613a as described in connection with mechanism 600a. Likewise, gear 603e includes cylindrical bushing 619e to provide a rotational bearing relationship between gear wheel 613a and shaft 655e.

Adjustment mechanism 600e has guide 625e including a channel 605e and rail 630e. Shaft 655e has mounting surface 667e that defines channel 605e. Channel 605e receives rail 630e therein, engaging mounting surface 637e of rail 630e with surface 667e. Channel 605e generally corresponds to a pathway represented by a line segment with the reference numeral 660e in FIG. 22. Rail 630e is fixed relative to block 22 using any suitable means that would occur to those skilled in the art.

In one preferred assembly process, rail 630e is mounted to establish a predetermined spatial relationship between pathway 660e and rotational center 54 of gear 50. Gear 603e is adjustably mounted to mesh with gears 50 and 200 by slidably engaging rail 630e in channel 605e. For any position of rail 630e in channel 605e, the spacing between rotational centers 54 and 650e is maintained to provide an acceptable minimal degree of backlash for the mesh between gears 603e and 50 in accordance with the predetermined spatial relationship between pathway 660e and rotational center 54. For the illustrated embodiment, pathway 660e is represented by a generally straight segment that is generally parallel to tangent line 610e; where line 610e is tangent to pitch circle 612e for gear 50 and pitch circle 614e for gear 603e. In an alternative embodiment, the pathway is represented by the dashed line in FIG. 22 with reference numeral 660e'; and is curved in the manner described in connection with pathway 660a' of mechanism 600a. The curvature of pathway 660e' is exaggerated in FIG. 22 to more clearly depict it.

Pathway 660e is also oriented to provide a substantially greater degree of spacing variation and corresponding backlash variation with respect to gear 200. Therefore, rail 630e may be moved along channel 605e to determine a position which minimizes backlash between gear 200 and gear 603e while not exceeding a desired level of backlash between gear 50 and gear 603e. Once a position is selected, shaft

655e is fixed relative to rail 630e and block 22 in a manner comparable to that described for plate 630a.

FIGS. 23 and 24 depict another idler gear adjustment mechanism 600f of the present invention. Mechanism 600f is illustrated as a substitute for idler gear 100 and mechanism 120 of system 20 with like reference numerals representing like features. Mechanism 600f includes gear 603f, mounting shaft 655f, and guide plate 630f that have been coupled to block 22 previously described in connection with system 20. Gear 603f includes gear wheel 613a described in connection with mechanism 600a. Likewise, gear 603f includes cylindrical bushing 619f to provide a rotational bearing relationship between gear wheel 613a and shaft 655f.

Shaft 655f has mounting surface 607f engaging a cheek 606f of plate 630f. Cheek 606f generally extends along a pathway represented by a line segment with the reference numeral 660f in FIG. 24. Plate 630f is fixed relative to block 22 using any suitable means that would occur to those skilled in the art.

In one preferred assembly process, plate 630f is mounted to establish a predetermined spatial relationship between pathway 660f and rotational center 54 of gear 50. Gear 603f is adjustably mounted to mesh with gears 50 and 200 by engaging mounting surface 607f of shaft 655f against cheek 606f in a sliding or rolling relationship. For any position of shaft 655f along cheek 606f, the spacing between rotational centers 54 and 650f is maintained to provide an acceptable minimal degree of backlash for the mesh between gears 603f and 50 in accordance with the predetermined spatial relationship between pathway 660f and rotational center 54. For the illustrated embodiment, pathway 660f is represented by a generally straight segment that is generally parallel to tangent line 610f; where line 610f is tangent to pitch circle 612f for gear 50 and pitch circle 614f for gear 603f. In an alternative embodiment, the pathway is represented by a dashed line in FIG. 24 with reference numeral 660f'; and is curved in the manner described in connection with pathway 660a' of mechanism 600a. The curvature of pathway 660f' is exaggerated in FIG. 24 to more clearly depict it.

Pathway 660f is also oriented to provide a substantially greater degree of spacing variation and corresponding backlash variation with respect to gear 200. Therefore, shaft 655f may be moved along cheek 606f to determine a position which minimizes backlash between gear 200 and gear 603f while not exceeding a desired level of backlash between gear 50 and gear 603f. Once a position is selected, shaft 655f is fixed relative to plate 630f and block 22 in a manner comparable to that described for plate 630a.

It should be understood that additional gears and different gear types may be used with mechanisms 600a, 600b, 600c, 600d, 600e, 600f (collectively designated mechanisms 600) as would occur to those skilled in the art. Moreover, various guide pathways may be established relative to center 204 instead of 54 with a corresponding change in backlash adjustability. Further, it should be appreciated that the various pins of mechanisms 600 may be joined by a bonding process, such as soldering, brazing, welding; by adhesive application; by threading, by a press-fit connection, or through such other means as would occur to those skilled in the art. The pins are preferably cylindrical in shape, but other embodiments may include pins with a different shape. Also various other guide members, such as slots, shoulders, walls, and channels, of mechanisms 600 may be machined, casted, or formed using techniques known to those skilled in the art. Preferably, all parts for mechanisms 600 are made from a metal or composite material suitable for heavy-duty diesel engine applications.

It should also be appreciated that each mechanism 600 facilitates assembly of a gear train with minimized backlash by constraining backlash variation between the idler gear and gear 54 to an acceptable level, and yet permitting generally independent adjustments to reduce backlash with gear 200. This feature makes it unnecessary to precision mount gear 200 and thus improve assembly effectiveness in terms of time and cost. Moreover, backlash reduction in accordance with the present invention generally provides a quieter, longer lasting gear train.

All publications, patents, and patent applications cited in this specification are herein incorporated by reference as if each individual publication, patent, or patent application was specifically and individually indicated to be incorporated by reference and set forth in its entirety herein.

While the invention has been illustrated and described in detail in the drawings and foregoing description, the same is to be considered as illustrative and not restrictive in character, it being understood that only the preferred embodiment has been shown and described and that all changes, modifications, and equivalents that come within the spirit of the invention as defined by the following claims are desired to be protected.

What is claimed is:

1. A gear train assembly for an engine, comprising:

- (a) a first scissor gear rotatably coupled to the engine, said first scissor gear including a first rotational center;
- (b) a second scissor gear rotatably coupled to the engine, said second scissor gear including a second rotational center; and
- (c) an adjustable idler gear mechanism coupled to the engine, said mechanism including an idler gear forming a first mesh with said first scissor gear and a second mesh with said second scissor gear, said idler gear including a third rotational center, said mechanism further including a first pin fixed relative to said third rotational center and a guide surface slidably engaged by said pin to define a pathway to adjustably position said third rotational center, said pathway defining a predetermined spatial relationship with said first rotational center to keep backlash between said first scissor gear and said idler gear at a predetermined minimum for a range of positions of said third rotational center along said pathway, said range of positions further providing a range of backlash adjustment between said idler gear and said second scissor gear.

2. The assembly of claim 1, wherein said pathway is generally straight.

3. The assembly of claim 1, wherein said pathway is generally curvilinear.

4. The assembly of claim 3, wherein said pathway is in the form of an arc that is equidistant from said first rotational center.

5. The assembly of claim 1, wherein said mechanism defines a slot bounded by said guide surface.

6. The assembly of claim 5, wherein said mechanism includes a second pin engaging said slot.

7. The assembly of claim 6, further comprising means for selectively fixing the second pin relative to said guide surface.

8. A gear train assembly for an engine, comprising:

- (a) a first scissor gear rotatably coupled to the engine, said first scissor gear including a first rotational center;
- (b) a second scissor gear rotatably coupled to the engine, said second scissor gear including a second rotational center; and

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(c) an adjustable idler gear mechanism coupled to the engine, said mechanism including an idler gear forming a first mesh with said first scissor gear and a second mesh with said second scissor gear, said idler gear including a third rotational center, said mechanism 5 including a channel and a rail slidably engaging said channel to adjust said third rotational center therealong, said rail defining a predetermined spatial relationship with said first rotational center to keep backlash 10 between said first scissor gear and said idler gear at a predetermined minimum for a range of positions of said

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third rotational center along said rail, said range of positions further providing a range of backlash adjustment between said idler gear and said second scissor gear.

9. The assembly of claim **8**, wherein said channel defines a generally straight pathway.

10. The assembly of claim **8**, further comprising means for fixing said rail in said channel.

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